The
AMERICAN SOCIETY OF HEATING
AND VENTILATING ENGINEERS

GUIDE

American Society *of* Heating *and* Ventilating Engineers

Guide, 1935

An Instrument of Service prepared for the Profession—and Containing reference data on the design and specification of heating and ventilating systems—Based on the Transactions—the Investigations of the Research Laboratory and Cooperating Institutions—and the Practice of the Members and Friends of the Society

together with a

Manufacturers' Catalog Data Section Containing Essential and Reliable Information Concerning Modern Equipment

also

The Roll of Membership of the Society

with

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PREFACE TO THE 13th EDITION

THE ambitious plans of the Guide Publication Committee, embodying several innovations to extend the usefulness of this reference volume, have been incorporated in this 13th annual edition of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE. The process of reviewing, revising and reconstructing the Technical Data Section and then coordinating the complex subject matter of the 41 chapters has engaged the attention of over 200 members so that THE A.S.H.V.E. GUIDE 1935 will appeal to an increasing number of readers and give them comprehensive data that are authoritative and practical.

Basic and fundamental data have been retained from previous editions and in those divisions where changes in practice have been observed modifications have been made in the text to bring the material up-to-date. The text of THE GUIDE 1935 now comprises two major divisions: the subject matter of chapters and a supplementary section of the problems and answers. These problems and their solutions presented as an appendix to each chapter represent the interpretation of the text by a competent engineer whose analysis has been carefully reviewed by the Guide Publication Committee. It should be understood, however, that for certain general questions, more than one answer can be made so that the addition of these questions which represent problems in practice greatly broadens the scope of THE GUIDE and generally enhances its usefulness. As developments in the manufacturing field have produced new apparatus and new applications of equipment for automatic heat and air conditioning to improve comfort, those chapters of THE GUIDE which discuss such equipment as controls, air washers, unit conditioners, oil burners, stokers, etc., have been reviewed by representative committees of engineers from manufacturers' associations so that the latest developments in their respective fields could be included.

The original conception of THE GUIDE outlined by its founders has been carefully safeguarded and the aim of the Guide Publication Committee is to have THE GUIDE 1935 maintain its leadership, and continue in its role, as the recognized authority in the fields of heating, ventilating and air conditioning. Thousands of engineers, architects, contractors and students have come under the influence of THE GUIDE since its first appearance in 1922 and they have found the data authoritative for their work in design, specification writing, installation or operation of apparatus and systems.

The Catalog Data of manufacturers is nearly 40 per cent greater in this current edition indicating that THE GUIDE is also recognized as an effective advertising medium for promoting the use of modern equipment.

THE GUIDE 1935 contains 150 pages more than the preceding volume and the Guide Publication Committee release this 13th edition of 10,000 copies, as a major contribution by the Society toward the general advancement of the engineering profession and its allied industries in the field of heating, ventilating and air conditioning.

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Special mention is due the several Committee members who acted as division chairmen and who devoted long hours and gave generously of their knowledge without thought of compensation other than the satisfaction of contributing to the advancement of the profession. The work of J. L. Blackshaw as technical assistant in the detailed work of compilation was worthy of special acknowledgment.

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CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

Chapter 1

FUNDAMENTALS OF HEATING AND AIR CONDITIONING

Dalton's Law, Dry- and Wet-Bulb Temperatures, Properties of Air, Humidity, Relative Humidity, Specific Humidity, Relation of Dew Point to Relative Humidity, Adiabatic Saturation of Air, Total Heat and Heat Content, Enthalpy, Psychrometric Chart, Properties of Steam, Properties of Water, Rate of Evaporation

AIR conditioning has for its objective the supplying and maintaining, in a room or other enclosure, of an atmosphere having a composition, temperature, humidity, and motion which will produce desired effects upon the occupants of the room or upon materials stored or handled in it.

Dry air is a mechanical mixture of gases composed, in percentage of volume, as follows[1]: nitrogen 78.03, oxygen 20.99, argon 0.94, carbon dioxide 0.03, and small amounts of hydrogen and other gases.

Atmospheric air at sea level is given in percentage by volume as: N_2 77.08, O_2 20.75, water vapor 1.2, A 0.93, CO_2 0.03 and H_2 0.01. The amount of water vapor varies greatly under different conditions and is frequently one of the most important constituents since it affects bodily comfort and greatly affects all kinds of hygroscopic materials.

LAW OF PARTIAL PRESSURES

A mixture of dry gases and water vapor, such as atmospheric air, obeys Dalton's Law of Partial Pressures: each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature had no other gas or vapor been present. If p = the observed pressure of the mixture and p_1, p_2, p_3, etc. = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3, \text{ etc.} \tag{1}$$

DRY- AND WET-BULB TEMPERATURES

Air is said to be saturated at a given temperature when the water vapor mixed with the air is in the dry saturated condition or, what is the equivalent, when the space occupied by the mixture holds the maximum possible weight of water *vapor* at that temperature. If the water vapor mixed with the dry air is superheated, *i.e.*, if its temperature is above the temperature of saturation for the actual water vapor partial pressure, the air is not saturated.

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[1] *International Critical Tables.*

The starting point of most applications of thermodynamic principles to air-conditioning problems is the experimental determination of the dry-bulb and wet-bulb temperatures, and sometimes the barometric pressure.

The *dry-bulb temperature* of the air is the temperature indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air. The *wet-bulb temperature* is determined by a thermometer with its bulb encased in a fine mesh fabric bag moistened with clean water and whirled through the air until the thermometer assumes a steady temperature. This steady temperature is the result of a dynamic equilibrium between the rate at which heat is transferred from the air to the water on the bulb and the rate at which this heat is utilized in evaporating moisture from the bulb. The rate at which heat is transferred from the air to the water is substantially proportional to the wet-bulb depression $(t - t')$, while the rate of heat utilization in evaporation is proportional to the difference between the saturation pressure of the water at the wet-bulb temperature and the actual partial pressure of the water vapor in the air $(e' - e)$. Carrier's equation for this dynamic equilibrium is

$$\frac{e' - e}{t - t'} = \frac{B - e'}{2800 - 1.3t'} \tag{2a}$$

In the form commonly used,

$$e = e' - \frac{(B - e')\,(t - t')}{2800 - 1.3t'} \tag{2b}$$

where

e = actual partial pressure of water vapor in the air, inches of mercury.
e' = saturation pressure at wet-bulb temperature, inches of mercury.
B = barometric pressure, inches of mercury.
t = dry-bulb temperature, degrees Fahrenheit.
t' = wet-bulb temperature, degrees Fahrenheit.

Formula 2b may be used to determine the actual partial pressure of the water vapor in a dry air-water vapor mixture. Then, from Dalton's Law of Partial Pressures, Equation 1, it follows that the partial pressure of the dry air is $(B - e)$.

If a mixture of dry air and water vapor, initially unsaturated, be cooled at constant pressure, the temperature at which condensation of the water vapor begins is called the *dew-point temperature*. Clearly the dew-point is the saturation temperature corresponding to the actual partial pressure, e, of the water vapor in the mixture.

PROPERTIES OF AIR

Density is variously defined as the mass per unit of volume, the weight per unit of volume, or the ratio of the mass, or weight, of a given volume of a substance to the mass, or weight, of an equal volume of some other substance such as water or air under standard conditions of temperature and pressure. The term *specific gravity* is more commonly used to express the latter relation but, when the gram is taken as the unit of mass and the cubic centimeter as the unit of volume, density and specific gravity have

the same meaning. The term *specific density* is sometimes used to distinguish the weight in pounds per cubic foot; and as here used, *density* is the weight in pounds of one cubic foot of a substance.

The density of air decreases with increase in temperature when under constant pressure. The density of dry air at 70 F and under standard atmospheric pressure (29.92 in. of Hg) is approximately 0.075 lb (see Table 1), while that of a mixture of air and saturated water vapor at the same temperature and barometric pressure is only about 0.0743 lb. In the mixture the density of the dry air is 0.0731 and that of the vapor is 0.00115 lb (see Table 2).

In order to make comparisons of air volumes or velocities it is necessary to reduce the observations to a common pressure and temperature basis. The basic pressure is usually taken as 29.92 in. of Hg, but no basic temperature is universally recognized. Common temperatures for this purpose are 32 F, 60 F, 68 F, and 70 F. Since 70 F is the most commonly specified temperature to which rooms for human occupancy must be heated, it is usually understood, when no other temperature is specified, that 70 F is the basic temperature for measuring the volume or the velocity of air in heating and ventilating work.

The *specific volume* of air is the volume in cubic feet occupied by one pound of the air. Under constant pressure the specific volume varies inversely as the density and directly as the absolute temperature.

The *specific heat* of *air* is the number of Btu required to raise the temperature of 1 lb of air 1 F. The specific heat at constant pressure, C_p, and that at constant volume, C_v, are different. The specific heat at constant pressure is commonly used and it varies, under a pressure of one atmosphere, from a minimum at about 32 F from which it *increases* with either increase or decrease of temperature. The value 0.24 is sufficiently accurate for use at ordinary temperatures, but the values range[1] from 0.2399 at 32 F to 0.2404 at 212 F, 0.2413 at 392 F, 0.243 at −108 F, and 0.252 at −301 F.

The *mean specific heat* of *water vapor* at constant pressure is taken as 0.45 for all general engineering computations.

Table 3 is intended to aid in determining the density of moist air, taking into account its temperature, pressure, and moisture content.

Example 1. To show the use of Table 3: Given air at 83 F dry-bulb and 68 F wet-bulb (or a depression of 15 deg) with a barometric pressure of 29.40 in. of mercury. What will be the weight of this air in pounds per cubic foot?

Solution. From Table 3 the weight of saturated air at 80 F and 29.00 in. barometer is found to be 0.07034 lb per cubic foot. There is a decrease of 0.00015 lb per degree dry-bulb temperature above 80 F. There is an increase of 0.00025 lb for each 0.1 in. above 29.00 in. From the last column of Table 3 it is found that there is an increase of approximately 0.000035 lb per degree wet-bulb depression when the dry-bulb is 83 F. Tabulating the items:

0.07034 = weight of saturated air at 80 F and 29.00 bar.
− 0.00045 = decrement for 3 deg dry-bulb, 3 × 0.00015.
+ 0.00100 = increment for 0.4 in. bar., 4 × 0.00025.
+ 0.00053 = increment for 15 deg wet-bulb depression, 15 × 0.000035.

0.07142 = weight in pounds per cubic foot of air at 83 F dry-bulb, 68 F wet-bulb, 29.40 in. bar.

Table 1. Properties of Dry Air[a]

Barometric Pressure 29.921 In.

Temperature Deg F	Weight per Cu Ft Pounds	Per Cent of Volume at 70 F	Btu Absorbed by One Cu Ft Dry Air per Deg F	Cu Ft Dry Air Warmed One Degree per Btu
0	0.08636	0.8680	0.02080	48.08
10	0.08453	0.8867	0.02039	49.05
20	0.08276	0.9057	0.01998	50.05
30	0.08107	0.9246	0.01957	51.10
40	0.07945	0.9434	0.01919	52.11
50	0.07788	0.9624	0.01881	53.17
60	0.07640	0.9811	0.01846	54.18
70	0.07495	1.0000	0.01812	55.19
80	0.07356	1.0190	0.01779	56.21
90	0.07222	1.0380	0.01747	57.25
100	0.07093	1.0570	0.01716	58.28
110	0.06968	1.0756	0.01687	59.28
120	0.06848	1.0945	0.01659	60.28
130	0.06732	1.1133	0.01631	61.32
140	0.06620	1.1320	0.01605	62.31
150	0.06510	1.1512	0.01578	63.37
160	0.06406	1.1700	0.01554	64.35
180	0.06205	1.2080	0.01506	66.40
200	0.06018	1.2455	0.01462	68.41
220	0.05840	1.2833	0.01419	70.48
240	0.05673	1.3212	0.01380	72.46
260	0.05516	1.3590	0.01343	74.46
280	0.05367	1.3967	0.01308	76.46
300	0.05225	1.4345	0.01274	78.50
350	0.04903	1.5288	0.01197	83.55
400	0.04618	1.6230	0.01130	88.50
450	0.04368	1.7177	0.01070	93.46
500	0.04138	1.8113	0.01018	98.24
550	0.03932	1.9060	0.00967	103.42
600	0.03746	2.0010	0.00923	108.35
700	0.03423	2.1900	0.00847	118.07
800	0.03151	2.3785	0.00782	127.88
900	0.02920	2.5670	0.00728	137.37
1000	0.02720	2.7560	0.00680	147.07

[a]From *Fan Engineering*.

It is usual to assume that dry air, moist air, and the water vapor in the air follow the laws of perfect gases. This assumption while not absolutely true, especially with saturated vapor at temperatures much above 140 F,

Table 2. Properties of Saturated Air[a]

Weights of Air, Vapor of Water, and Saturated Mixture of Air and Vapor at 29.921 Inches of Mercury

Temp. Deg F	Weight in a Cubic Foot of Mixture			Btu Absorbed by One Cubic Foot Sat. Air per Deg F	Cubic Feet Sat. Air Warmed One Degree per Btu	Specific Heat Btu per Pound of Mixture
	Weight of Dry Air Pounds	Weight of Vapor Pounds	Total Weight of the Mixture Pounds			
0	0.08625	0.000068	0.08632	0.02083	48.02	0.2413
10	0.08433	0.000110	0.08444	0.02039	49.05	0.2415
20	0.08246	0.000176	0.08264	0.01998	50.07	0.2418
30	0.08062	0.000277	0.08090	0.01958	51.07	0.2420
40	0.07878	0.000409	0.07919	0.01921	52.06	0.2426
50	0.07694	0.000587	0.07753	0.01885	53.05	0.2431
60	0.07506	0.000828	0.07589	0.01851	54.02	0.2439
70	0.07310	0.001151	0.07425	0.01819	54.97	0.2450
80	0.07103	0.001578	0.07261	0.01790	55.87	0.2465
90	0.06879	0.002134	0.07092	0.01762	56.76	0.2485
100	0.06635	0.002850	0.06920	0.01736	57.59	0.2509
110	0.06364	0.003762	0.06740	0.01714	58.35	0.2543
120	0.06060	0.004914	0.06551	0.01695	59.00	0.2587
130	0.05715	0.006351	0.06350	0.01679	59.56	0.2644
140	0.05319	0.008120	0.06131	0.01668	59.96	0.2721
150	0.04864	0.010295	0.05894	0.01662	60.17	0.2820
160	0.04340	0.012936	0.05634	0.01662	60.17	0.2950
170	0.03734	0.016108	0.05345	0.01668	59.96	0.3121
180	0.03035	0.019896	0.05025	0.01684	59.38	0.3351
190	0.02228	0.024400	0.04668	0.01710	58.49	0.3663
200	0.01300	0.029715	0.04272	0.01749	57.18	0.4094
210	0.00230	0.035938	0.03824	0.01802	55.50	0.4712
212	0.00000	0.037307	0.03731	0.01815	55.10	0.4865

[a]From *Fan Engineering*.

is sufficiently accurate for practical purposes and it greatly simplifies computations.

Boyle's Law refers to the relation between the pressure and volume of a gas, and may be stated as follows: *With temperature constant, the volume of a given weight of gas varies inversely as its absolute pressure.* Hence, if P_1 and P_2 represent the initial and final absolute pressures, and V_1 and V_2 represent corresponding volumes of the same mass, say one pound of gas, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$, or $P_1 V_1 = P_2 V_2$, but since $P_1 V_1$ for any given case is a definite constant quantity, it follows that the product of the absolute

TABLE 3. WEIGHTS OF SATURATED AND PARTLY SATURATED AIR FOR VARIOUS BAROMETRIC AND HYGROMETRIC CONDITIONS[a, b]

Pounds per Cubic Foot

Dry-Bulb Temperature Deg F	Barometric Pressure—Inches															Approx. Average Increase in Weight per Deg Wet-Bulb Depression
	26			27			28			29			30			
	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar.	
0	0.07500	0.00016	0.00029	0.07788	0.00016	0.00029	0.08077	0.00017	0.00029	0.08365	0.00018	0.00029	0.08654	0.00019	0.00029	0.000015
10	0.07338	0.00016	0.00028	0.07620	0.00016	0.00028	0.07903	0.00017	0.00028	0.08185	0.00018	0.00028	0.08468	0.00018	0.00028	0.000016
20	0.07180	0.00016	0.00028	0.07456	0.00016	0.00028	0.07733	0.00017	0.00028	0.08009	0.00018	0.00028	0.08286	0.00018	0.00028	0.000017
30	0.07027	0.00015	0.00027	0.07297	0.00016	0.00027	0.07569	0.00016	0.00027	0.07839	0.00017	0.00027	0.08110	0.00017	0.00027	0.000019
40	0.06879	0.00015	0.00026	0.07143	0.00015	0.00027	0.07409	0.00016	0.00027	0.07675	0.00016	0.00027	0.07942	0.00017	0.00027	0.000021
50	0.06732	0.00015	0.00026	0.06992	0.00015	0.00026	0.07252	0.00016	0.00026	0.07512	0.00016	0.00026	0.07773	0.00016	0.00026	0.000023
60	0.06588	0.00015	0.00026	0.06843	0.00015	0.00026	0.07098	0.00015	0.00026	0.07353	0.00016	0.00026	0.07609	0.00016	0.00026	0.000026
70	0.06442	0.00015	0.00025	0.06692	0.00015	0.00025	0.06943	0.00015	0.00025	0.07193	0.00016	0.00025	0.07440	0.00016	0.00025	0.000029
80	0.06297	0.00015	0.00025	0.06542	0.00015	0.00025	0.06789	0.00015	0.00025	0.07034	0.00015	0.00025	0.07280	0.00016	0.00025	0.000034
90	0.06146	0.00015	0.00024	0.06388	0.00016	0.00024	0.06629	0.00016	0.00024	0.06870	0.00016	0.00024	0.07112	0.00017	0.00024	0.000039
100	0.05991	0.00016	0.00024	0.06228	0.00016	0.00024	0.06465	0.00016	0.00024	0.06703	0.00017	0.00024	0.06939	0.00018	0.00024	0.000044
110	0.05828	0.00016	0.00023	0.06060	0.00017	0.00023	0.06293	0.00017	0.00023	0.06526	0.00018	0.00023	0.06759	0.00019	0.00023	0.000051
120	0.05653	0.00018	0.00023	0.05882	0.00018	0.00023	0.06111	0.00018	0.00023	0.06339	0.00019	0.00023	0.06569	0.00020	0.00023	0.000059
130	0.05467	0.00019	0.00023	0.05692	0.00019	0.00023	0.05917	0.00019	0.00023	0.06142	0.00020	0.00023	0.06367	0.00022	0.00023	0.000068
140	0.05262	0.00021	0.00022	0.05483	0.00021	0.00022	0.05704	0.00021	0.00022	0.05925	0.00022	0.00022	0.06147	0.00024	0.00022	0.000078
150	0.05036	0.00023	0.00022	0.05253	0.00023	0.00022	0.05471	0.00023	0.00022	0.05689	0.00024	0.00022	0.05906	0.00026	0.00022	0.000090
160	0.04788	0.00025	0.00022	0.05001	0.00025	0.00022	0.05216	0.00026	0.00021	0.05430	0.00026	0.00021	0.05644	0.00029	0.00021	0.000103
170	0.04509	0.00028	0.00021	0.04720	0.00028	0.00021	0.04931	0.00029	0.00021	0.05141	0.00031	0.00021	0.05352	0.00033	0.00021	0.000118
180	0.04197	0.00031	0.00021	0.04404	0.00031	0.00021	0.04611	0.00032	0.00021	0.04818	0.00034	0.00021	0.05026	0.00036	0.00021	0.000134
190	0.03845	0.00035	0.00021	0.04049	0.00036	0.00021	0.04253	0.00036	0.00021	0.04457	0.00037	0.00021	0.04662	0.00038	0.00021	0.000153
200	0.03449	0.00040	0.00020	0.03650	0.00040	0.00020	0.03851	0.00040	0.00020	0.04052	0.00041	0.00020	0.04254	0.00041	0.00020	0.000173

[a]From *Fan Engineering*.

[b]A convenient and accurate chart for quickly determining the weight of air under any condition of dry-bulb, wet-bulb, and pressure is *A Chart for Determining the Weight of Moist Air in Pounds per Cubic Foot*, by John E. Younger. Published in *Mechanical Engineering*, June, 1925.

pressure and volume of a gas is a constant, or $PV = C$, when T is kept constant. Any change in the pressure and volume of a gas at constant temperature is called an *isothermal change.*

Charles' Law refers to the relation among pressure, volume, and temperature of a gas and may be stated as follows: *The volume of a given weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume.* Hence, when heat is added at constant volume, V_c, the resulting equation is $\frac{P_2}{P_1} = \frac{T_2}{T_1}$, or, for the same temperature range at constant pressure, P_c, the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general, for any weight of gas, W, since volume is proportional to weight, the relation among P, V, and T is

$$PV = WRT \tag{3}$$

where

P = the absolute pressure of the gas, pounds per square foot.
V = the volume of the weight W, cubic feet.
W = the weight of the gas, pounds.
R = a constant depending on the nature of the gas. The average value of R for air is 53.34.
T = the absolute temperature, degrees Fahrenheit.

This is the characteristic equation for a perfect gas, and while no gases are perfect in this sense, they conform so nearly that Equation 3 will apply to most engineering computations.

HUMIDITY

Humidity is the water vapor mixed with dry air in the atmosphere. *Absolute humidity* has a multiplicity of meanings, but usually the term refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. With this meaning, absolute humidity is nothing but the actual density of the water vapor in the mixture and might better be so called. A study of Keenan's Steam Tables[2] indicates that water vapor, either saturated or super-heated, at partial pressures lower than 4 in. of mercury may be treated as a gas with a gas constant R of 1.21 in the characteristic equation of the gas $pV = wR\ (t + 460)$. Within such limits, the density (δ) of water vapor is

$$\delta = \frac{w}{V} = \frac{e}{1.21\ (t + 460)} \text{ (pounds per cubic foot)} \tag{4a}$$

$$= \frac{5785\ e}{t + 460} \text{ (grains per cubic foot)} \tag{4b}$$

where

e = actual partial pressure of vapor, inches of mercury.
t = dry-bulb temperature, degrees Fahrenheit.

[2]Published by *American Society of Mechanical Engineers*, see abstract in Table 7.

Specific Humidity

It simplifies many problems which deal with mixtures of dry air and water vapor to express the weight or the mass of the vapor in terms of the weight or the mass of dry air. If the weight of the water vapor in a mixture be divided by the weight of the dry air, and the weight of dry air be made unity, we have an expression of the weight of water vapor carried by a unit weight of dry air. This relation has no generally accepted name. It has been variously called: mixing ratio, proportionate humidity, mass or density ratio, absolute humidity, and specific humidity. Of all these terms *specific humidity* is the most suggestive of the meaning which it is desired to express and it has found considerable use in this sense even though it is defined in International Critical Tables as the ratio of the mass of vapor to the total mass. It will be understood here that *specific humidity* refers to the weight of water vapor in pounds carried by one pound of dry air.

The gas constant for dry air, when the partial pressure of the air is expressed in inches of Hg, is 0.753; so that the specific humidity, if represented by W, is

$$W = \frac{e}{1.21\ (t + 460)} \div \frac{B\text{-}e}{0.753\ (t + 460)}$$

$$= 0.622 \left(\frac{e}{B\text{-}e}\right) \text{(pounds)} \quad (5a)$$

$$= 4354 \left(\frac{e}{B\text{-}e}\right) \text{(grains)} \quad (5b)$$

where

e = actual partial pressure of vapor, inches of mercury.
B = total pressure of mixture (barometric pressure), inches of mercury.

Relative Humidity

Relative humidity (Φ) is either the ratio of the actual partial pressure, e, of the water vapor in the air to the saturation pressure, e_t, at the dry-bulb temperature, or the ratio of the actual density, δ, of the vapor to the density of saturated vapor, δ_t, at the dry-bulb temperature. That is:

$$\Phi = \frac{e}{e_t} = \frac{\delta}{\delta_t} \quad (6)$$

The relative humidity of a given mixture at a given temperature is not the same as the specific humidity, W, of the mixture divided by the specific humidity, W_t, of saturated vapor at the same temperature, for from Equations 5a and 6

$$\frac{W}{W_t} = 0.622 \left(\frac{\Phi\ e_t}{P - \Phi\ e_t}\right) \div 0.622 \left(\frac{e_t}{B\text{-}e_t}\right) = \frac{\Phi\ (B\text{-}e_t)}{B - \Phi e_t} \quad (7)$$

The specific humidity of an unsaturated air-vapor mixture cannot, therefore, be accurately found by multiplying the specific humidity of saturated vapor by its relative humidity; although the error is usually small especially when the relative humidity is high.

With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and

dew-point temperatures are equal. With a relative humidity less than 100 per cent, the dry-bulb exceeds the wet-bulb, and the wet-bulb exceeds the dew-point temperature.

RELATION OF DEW POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew point and the relative humidity and this is found most useful in air conditioning work. This relationship is, that for a fixed relative humidity there is substantially a constant difference between the dew point and the dry-bulb temperature over a considerable temperature range. Table 4, giving the dry-bulb and dew-point temperatures and the dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

TABLE 4. DRY-BULB AND DEW-POINT TEMPERATURES FOR 50 PER CENT RELATIVE HUMIDITY

Dry-bulb temperature	65.0	70.0	75.0	80.0	85.0	90.0
Dew-point temperature	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew-point and dry-bulb temperature	19.2	19.5	19.75	20.25	20.75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_1}{20}}} \tag{8}$$

where

t_1 = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills and tobacco factories, where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and the dew-point temperature, to control the relative humidity in the room.

Table 5 gives, for different temperatures, the density of saturated vapor,

TABLE 5. MIXTURES OF AIR AND SATURATED WATER VAPOR[a]

Temp., F	[b]Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft		Heat Content in Btu of 1 Lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[c]Heat Content in Btu of 1 Lb of Dry Air with Vapor to Saturate It
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
0	0.0375	0.0184	0.0000674	0.472	0.000781	5.47	11.58	11.59	0.0	0.852	0.852
2	.0417	.0204	.0000746	.522	.000869	6.08	11.63	11.65	0.482	0.946	1.428
4	.0462	.0227	.0000823	.576	.000963	6.74	11.68	11.70	0.964	1.047	2.011
6	.0512	.0252	.0000909	.636	.001067	7.47	11.73	11.75	1.446	1.159	2.605
8	.0567	.0279	.0001001	.701	.001183	8.28	11.78	11.80	1.928	1.285	3.213
10	0.0628	0.0308	0.0001103	0.772	0.001309	9.16	11.83	11.86	2.411	1.420	3.831
12	.0694	.0341	.000121	.850	.001447	10.13	11.88	11.91	2.893	1.568	4.461
14	.0766	.0376	.000134	.935	.001599	11.19	11.94	11.97	3.375	1.731	5.106
16	.0846	.0415	.000147	1.028	.001764	12.35	11.99	12.02	3.858	1.908	5.766
18	.0932	.0458	.000161	1.128	.001946	13.62	12.04	12.08	4.340	2.103	6.443
20	0.1027	0.0504	0.000177	1.237	0.002144	15.01	12.09	12.13	4.823	2.314	7.137
22	.1130	.0555	.000194	1.356	.002360	16.52	12.14	12.19	5.305	2.545	7.850
24	.1242	.0610	.000212	1.485	.002596	18.17	12.19	12.24	5.787	2.796	8.583
26	.1365	.0670	.000232	1.625	.002854	19.98	12.24	12.30	6.270	3.071	9.341
28	.1499	.0736	.000254	1.776	.003134	21.94	12.29	12.35	6.752	3.370	10.122
30	0.1646	0.0809	0.000278	1.943	0.003444	24.11	12.34	12.41	7.234	3.699	10.933
32	.1806	.0887	.000303	2.124	.003782	26.47	12.39	12.47	7.716	4.058	11.783
33	.1880	.0923	.000315	2.206	.003938	27.57	12.41	12.49	7.96	4.22	12.18
34	.1957	.0961	.000327	2.292	.004100	28.70	12.44	12.52	8.20	4.40	12.60
35	0.2036	0.1000	0.000340	2.380	0.004268	29.88	12.47	12.55	8.44	4.57	13.02
36	.2119	.1041	.000353	2.471	.004442	31.09	12.49	12.58	8.68	4.76	13.44
37	.2204	.1083	.000367	2.566	.004622	32.35	12.52	12.61	8.93	4.95	13.87
38	.2292	.1126	.000381	2.663	.004809	33.66	12.54	12.64	9.17	5.14	14.31
39	.2384	.1171	.000395	2.764	.005002	35.01	12.57	12.67	9.41	5.35	14.76
40	0.2478	0.1217	0.000410	2.868	0.005202	36.41	12.59	12.70	9.65	5.56	15.21
41	.2576	.1266	.000425	2.976	.005410	37.87	12.62	12.73	9.89	5.78	15.67
42	.2678	.1315	.000441	3.087	.005625	39.38	12.64	12.76	10.14	6.01	16.14
43	.2783	.1367	.000457	3.201	.005848	40.93	12.67	12.79	10.38	6.24	16.62
44	.2891	.1420	.000474	3.319	.006078	42.55	12.69	12.82	10.62	6.48	17.10

[a]Based on *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Below 32 F the pressure of saturated vapor in contact with ice is given.
[c]Values in this column do not include the heat of the liquid.

TABLE 5. MIXTURES OF AIR AND SATURATED WATER VAPOR[a] (CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft		Heat Content in Btu of 1 lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[b]Heat Content in Btu of 1 lb of Dry Air with Vapor to Saturate it
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
45	0.3003	0.1475	0.000492	3.442	0.00632	44.21	12.72	12.85	10.86	6.73	17.59
46	.3120	.1532	.000510	3.568	.00656	45.94	12.74	12.88	11.10	6.99	18.09
47	.3240	.1591	.000528	3.698	.00682	47.73	12.77	12.91	11.34	7.26	18.60
48	.3364	.1652	.000547	3.832	.00708	49.58	12.79	12.94	11.58	7.54	19.12
49	.3492	.1715	.000567	3.970	.00736	51.49	12.82	12.97	11.83	7.83	19.65
50	0.3624	0.1780	0.000588	4.113	0.00764	53.47	12.84	13.00	12.07	8.12	20.19
51	.3761	.1848	.000609	4.260	.00793	55.52	12.87	13.03	12.31	8.43	20.74
52	.3903	.1917	.000630	4.411	.00823	57.64	12.89	13.07	12.55	8.75	21.30
53	.4049	.1989	.000653	4.568	.00855	59.83	12.92	13.10	12.79	9.08	21.87
54	.4200	.2063	.000676	4.729	.00887	62.09	12.95	13.13	13.03	9.41	22.45
55	0.4356	0.2140	0.000699	4.895	0.00920	64.43	12.97	13.16	13.28	9.76	23.04
56	.4517	.2219	.000724	5.066	.00955	66.85	13.00	13.20	13.52	10.13	23.64
57	.4684	.2300	.000749	5.242	.00991	69.35	13.02	13.23	13.76	10.50	24.25
58	.4855	.2384	.000775	5.424	.01028	71.93	13.05	13.26	14.00	10.89	24.88
59	.5032	.2471	.000802	5.611	.01066	74.60	13.07	13.30	14.24	11.28	25.52
60	0.5214	0.2561	0.000829	5.804	0.01105	77.3	13.10	13.33	14.48	11.69	26.18
61	.5403	.2654	.000858	6.003	.01146	80.2	13.12	13.36	14.72	12.12	26.84
62	.5597	.2749	.000887	6.208	.01188	83.2	13.15	13.40	14.97	12.56	27.52
63	.5798	.2848	.000917	6.418	.01231	86.2	13.17	13.43	15.21	13.01	28.22
64	.6005	.2949	.000948	6.633	.01276	89.3	13.20	13.47	15.45	13.48	28.93
65	0.6218	0.3054	0.000979	6.855	0.01323	92.6	13.22	13.50	15.69	13.96	29.65
66	.6438	.3162	.001012	7.084	.01370	95.9	13.25	13.54	15.93	14.46	30.39
67	.6664	.3273	.001046	7.320	.01420	99.4	13.27	13.58	16.18	14.97	31.15
68	.6898	.3388	.001080	7.563	.01471	103.0	13.30	13.61	16.42	15.50	31.92
69	.7139	.3506	.001116	7.813	.01524	106.6	13.32	13.65	16.66	16.05	32.71
70	0.7386	0.3628	0.001153	8.069	0.01578	110.5	13.35	13.69	16.90	16.61	33.51
71	.7642	.3754	.001190	8.332	.01634	114.4	13.38	13.73	17.14	17.19	34.33
72	.7906	.3883	.001229	8.603	.01692	118.4	13.40	13.76	17.38	17.79	35.17
73	.8177	.4016	.001269	8.882	.01751	122.6	13.43	13.80	17.63	18.41	36.03
74	.8456	.4153	.001310	9.168	.01813	126.9	13.45	13.84	17.87	19.05	36.91

[a]Reprinted by permission from *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Values in this column do not include the heat of the liquid.

TABLE 5. MIXTURES OF AIR AND SATURATED WATER VAPOR[a] (CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft		Heat Content in Btu of 1 Lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[b]Heat Content in Btu of 1 Lb of Dry Air with Vapor to Saturate It
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
75	0.8744	0.4295	0.001352	9.46	0.01877	131.4	13.48	13.88	18.11	19.71	37.81
76	.9040	.4440	.001395	9.76	.01942	135.9	13.50	13.92	18.35	20.38	38.73
77	.9345	.4590	.001439	10.07	.02010	140.7	13.53	13.96	18.59	21.08	39.67
78	.9658	.4744	.001485	10.39	.02080	145.6	13.55	14.00	18.84	21.80	40.64
79	.9981	.4903	.001532	10.72	.02152	150.6	13.58	14.05	19.08	22.55	41.63
80	1.0314	0.5066	0.001580	11.06	0.02226	155.8	13.60	14.09	19.32	23.31	42.64
81	1.0656	.5234	.001629	11.40	.02303	161.2	13.63	14.13	19.56	24.11	43.67
82	1.1008	.5406	.001680	11.76	.02381	166.7	13.65	14.17	19.80	24.92	44.72
83	1.1370	.5584	.001732	12.12	.02463	172.4	13.68	14.22	20.04	25.76	45.80
84	1.174	.5767	.001786	12.50	.02547	178.3	13.70	14.26	20.29	26.62	46.91
85	1.212	0.5955	0.001841	12.89	0.02634	184.4	13.73	14.31	20.53	27.51	48.04
86	1.251	.6148	.001897	13.28	.02723	190.6	13.75	14.35	20.77	28.43	49.20
87	1.292	.6347	.001955	13.68	.02815	197.0	13.78	14.40	21.01	29.38	50.39
88	1.334	.6551	.002014	14.10	.02910	203.7	13.80	14.45	21.25	30.35	51.61
89	1.377	.6761	.002075	14.53	.03008	210.6	13.83	14.50	21.50	31.36	52.86
90	1.421	0.6977	0.002137	14.96	0.03109	217.6	13.86	14.55	21.74	32.39	54.13
91	1.466	.7200	.002201	15.41	.03213	224.9	13.88	14.60	21.98	33.46	55.44
92	1.512	.7427	.002267	15.87	.03320	232.4	13.91	14.65	22.22	34.59	56.78
93	1.560	.7660	.002334	16.34	.03430	240.1	13.93	14.70	22.46	35.69	58.15
94	1.609	.7901	.002403	16.82	.03544	247.1	13.96	14.75	22.71	36.86	59.56
95	1.659	0.8148	0.002474	17.32	0.03662	256.3	13.98	14.80	22.95	38.06	61.01
96	1.710	.8401	.002546	17.82	.03783	264.8	14.01	14.86	23.19	39.30	62.48
97	1.763	.8662	.002621	18.35	.03908	273.6	14.03	14.91	23.43	40.57	64.00
98	1.818	.8929	.002697	18.88	.04036	282.5	14.06	14.97	23.67	41.88	65.55
99	1.874	.9204	.002775	19.42	.04169	291.8	14.08	15.02	23.91	43.24	67.15
100	1.931	0.9486	0.002855	19.98	0.04305	301.3	14.11	15.08	24.16	44.63	68.79
101	1.990	0.9775	.002937	20.56	.04446	311.2	14.14	15.14	24.40	46.07	70.47
102	2.051	1.0072	.003021	21.15	.04591	321.4	14.16	15.20	24.64	47.54	72.18
103	2.113	1.0376	.003107	21.75	.04741	331.9	14.19	15.26	24.88	49.07	73.95
104	2.176	1.0689	.003195	22.36	.04895	342.7	14.21	15.33	25.13	50.64	75.77

[a]Reprinted by permission from *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Values in this column do not include the heat of the liquid.

TABLE 5. MIXTURES OF AIR AND SATURATED WATER VAPOR[a] (CONTINUED)

TEMP., F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR				VOLUME IN CU FT		HEAT CONTENT IN BTU OF 1 LB OF DRY AIR ABOVE 0 F	LATENT HEAT OF VAPOR, BTU	[b]HEAT CONTENT IN BTU OF 1 LB OF DRY AIR WITH VAPOR TO SATURATE IT
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
105	2.241	1.1010	0.003285	22.99	0.0505	354	14.24	15.39	25.37	52.26	77.63
106	2.308	1.134	.003377	23.64	.0522	365	14.26	15.46	25.61	53.92	79.53
107	2.377	1.168	.003472	24.30	.0539	377	14.29	15.52	25.85	55.64	81.49
108	2.448	1.202	.003568	24.98	.0556	389	14.31	15.59	26.09	57.41	83.50
109	2.520	1.238	.003667	25.67	.0574	402	14.34	15.66	26.33	59.23	85.57
110	2.594	1.274	0.003769	26.38	0.0593	415	14.36	15.73	26.58	61.11	87.69
111	2.670	1.311	.003873	27.11	.0612	428	14.39	15.80	26.82	63.04	89.86
112	2.748	1.350	.003979	27.85	.0631	442	14.41	15.87	27.06	65.04	92.10
113	2.827	1.389	.004087	28.61	.0652	456	14.44	15.95	27.30	67.10	94.40
114	2.909	1.429	.004198	29.39	.0673	471	14.46	16.02	27.55	69.22	96.77
115	2.993	1.470	0.004312	30.18	0.0694	486	14.49	16.10	27.79	71.40	99.10
116	3.079	1.512	.004428	31.00	.0717	502	14.52	16.18	28.03	73.65	101.68
117	3.167	1.555	.004547	31.83	.0739	518	14.54	16.26	28.27	75.97	104.24
118	3.257	1.600	.004669	32.68	.0763	534	14.57	16.35	28.51	78.36	106.87
119	3.349	1.645	.004793	33.55	.0788	551	14.59	16.43	28.76	80.80	109.56
120	3.444	1.692	0.004920	34.44	0.0813	569	14.62	16.52	29.00	83.37	112.37
125	3.952	1.941	.005599	39.19	.0953	667	14.75	16.99	30.21	97.33	127.54
130	4.523	2.221	.006356	44.49	.1114	780	14.88	17.53	31.42	113.64	145.06
135	5.163	2.536	.007197	50.38	.1305	913	15.00	18.13	32.63	132.71	165.34
140	5.878	2.887	.008130	56.91	.1532	1072	15.13	18.84	33.85	155.37	189.22
145	6.677	3.280	0.00916	64.1	0.1800	1260	15.26	19.64	35.06	182.05	217.1
150	7.566	3.716	.01030	72.1	.2122	1485	15.39	20.60	36.27	214.03	250.3
155	8.554	4.201	.01156	80.9	.2511	1758	15.52	21.73	37.48	252.61	290.1
160	9.649	4.739	.01294	90.6	.2987	2091	15.64	23.09	38.69	299.55	338.2
165	10.860	5.334	.01445	101.1	.3577	2504	15.77	24.75	39.91	357.75	397.7
170	12.20	5.990	0.01611	112.8	0.4324	....	15.90	26.84	41.12	431.2	472.3
175	13.67	6.71	.01793	125.5	.5290	....	16.03	29.51	42.33	526.0	568.3
180	15.29	7.51	.01991	139.4	.6577	....	16.16	33.04	43.55	651.9	695.5
185	17.07	8.38	.02206	154.4	.8359	....	16.28	37.89	44.76	826.1	870.9
190	19.01	9.34	.02441	170.9	1.0985	....	16.41	45.00	45.97	1082.3	1128.3
200	23.46	11.53	0.02972	208.0	2.2953	....	16.67	77.24	48.40	2247.5	2296

[a]Reprinted by permission from *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Values in this column do not include the heat of the liquid.

δ_t, the weight of saturated vapor mixed with 1 lb of dry air, W_t, (at a relative humidity of 100 per cent and a barometric pressure, B, of 29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 lb of dry air (at a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The preceding equations or the data from Table 5 may be conveniently used in solving the following typical problems: (See Table 6 for temperatures below 0F.)

Example 2. Humidifying and Heating. Air is to be maintained at 70 F with a relative humidity of 40 per cent ($\Phi = 0.4$) when the outside air is at 0 F and 70 per cent relative humidity ($\Phi = 0.7$) and a barometric pressure, B, of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Equation 5a and Table 5,

$$W_1 = 0.622\left(\frac{0.7 \times 0.0375}{29.92 - 0.0263}\right) = 0.000547 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622\left(\frac{0.4 \times 0.7386}{29.92 - 0.295}\right) = 0.00618 \text{ lb per pound of dry air.}$$

The water vapor added per pound of dry air must be ($W_2 - W_1$) or 0.005633 lb. By inspection of Table 5, W_t = 0.00618 at 44.5 F, so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 5 is

$W_1 = 0.7 \times 0.000781 = 0.000547$ lb per pound of dry air.
$W_2 = 0.4 \times 0.01578 = 0.006312$ lb per pound of dry air.

The water vapor added per pound of dry air is approximately 0.005765 lb and the dew-point temperature is approximately 45 F. The degree of approximation is evident.

Example 3. Dehumidifying and Cooling. Air with a dry-bulb temperature of 84 F, a wet-bulb of 70 F, or a relative humidity of 50 per cent ($\Phi = 0.5$), and a barometric pressure, B, of 29.92 in. of mercury is to be cooled to 54 F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Equation 5a and Table 5,

$$W_1 = 0.622\left(\frac{0.5 \times 1.174}{29.92 - 0.587}\right) = 0.01245 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622\left(\frac{0.42}{29.92 - 0.42}\right) = 0.00887 \text{ lb per pound of dry air.}$$

Since $W_1 = W_t$ when $t = 63.3$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is ($W_1 - W_2$) or 0.00358 lb per pound of dry air.

An approximate result is

$W_1 = 0.5 \times 0.02547 = 0.01274$ lb per pound of dry air.
$W_2 = 1 \times 0.00887 = 0.00887$ lb per pound of dry air, since the exit air is saturated.

Since $W_1 = W_t$ at $t = 64$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.00387 lb per pound of dry air. The degree of approximation is again evident.

ADIABATIC SATURATION OF AIR

The process of adiabatic saturation of air is of considerable importance in air conditioning. Suppose that 1 lb of dry air, initially unsaturated but carrying W lb of water vapor with a dry-bulb temperature, t, and a wet-

bulb temperature, t', be made to pass through a tunnel containing an exposed water surface. Further assume the tunnel to be completely insulated, thermally, so that the only heat transfer possible is that between the air and water. As the air passes over the water surface, it will gradually pick up water vapor and will approach saturation *at the initial wet-bulb temperature of the air*, if the water be supplied at this wet-bulb temperature. During the process of adiabatic saturation, then, the dry-bulb temperature of the air drops to the wet-bulb temperature as a limit, the wet-bulb temperature remains substantially constant, and the weight of water vapor associated with each pound of dry air increases to $W_{t'}$, as a limit, *where* $W_{t'}$ is the weight of saturated vapor per pound of dry air for saturation at the wet-bulb temperature.

Example 4. If air with a dry-bulb of 85 F and a wet-bulb of 70 F be saturated adiabatically by spraying with recirculated water, what will be the final temperature and the vapor content of the air?

Solution. The final temperature will be equal to the initial wet-bulb temperature or 70 F, and since the air is saturated at this temperature, from Table 5, $W = 0.01578$ lb per pound of dry air.

In the adiabatic saturation process, since the heat given up by the dry air and associated vapor in cooling to the wet-bulb temperature is utilized in evaporation of water at the wet-bulb temperature, W. H. Carrier has pointed out[3] that the equation for the process of *adiabatic saturation*, and hence for a process of *constant wet-bulb temperature*, is:

$$h'_{fg}\,(W_{t'} - W) = c_{p_a}\,(t - t') + c_{p_s} W\,(t - t') \qquad (9a)$$

and using $c_{p_a} = 0.24$ and $c_{p_s} = 0.45$

$$h'_{fg}\,(W_{t'} - W) = (0.24 + 0.45W)\,(t - t') \qquad (9b)$$

where

h'_{fg} = latent heat of vaporization at t', Btu per pound.

$(W_{t'} - W)$ = increase in vapor associated with 1 lb of dry air when it is saturated adiabatically from an initial dry-bulb temperature, t, and an initial vapor content, W, pounds.

Knowing any two of the three primary variables, t, t', or W, the third may be found from this equation for any process of adiabatic saturation.

TOTAL HEAT AND HEAT CONTENT

The total heat of a mixture of dry air and water vapor was originally defined by W. H. Carrier as

$$\Sigma = c_{p_a}\,(t - 0) + W\,[h'_{fg} + c_{p_s}\,(t - t')] \qquad (10)$$

where

Σ = total heat of the mixture, Btu per pound of dry air.
c_{p_a} = mean specific heat at constant pressure of dry air.
c_{p_s} = mean specific heat at constant pressure of water vapor.
t = dry-bulb temperature, degrees Fahrenheit.
t' = wet-bulb temperature, degrees Fahrenheit.

[3] *A.S.M.E. Transactions*, Vol. 33, 1911, p. 1005.

TABLE 6. PROPERTIES OF SATURATED WATER VAPOR AT LOW TEMPERATURES[a]

Barometer, 29.92 Inches of Mercury

Temperature F	Vapor Pressure In. Hg × 10⁻⁵	Weight of Saturated Vapor per Lb Dry Air × 10⁻⁵	Btu per Lb of Vapor (32 F Datum)	Temperature F	Vapor Pressure In. Hg × 10⁻⁵	Weight of Saturated Vapor per Lb Dry Air × 10⁻⁵	Btu per Lb of Vapor (32 F Datum)
−130	0.276	0.005738	1000.7	−85	15.87	0.3299	1021.0
−129	.306	.006362	1001.2	−84	17.20	.3576	1021.4
−128	.338	.007027	1001.6	−83	18.58	.3863	1021.9
−127	.373	.007755	1002.1	−82	20.10	.4179	1022.3
−126	.411	.008545	1002.5	−81	21.72	.4516	1022.8
−125	.455	.009459	1003.0	−80	23.47	.4879	1023.2
−124	.499	.01037	1003.4	−79	25.34	.5268	1023.7
−123	.542	.01127	1003.9	−78	27.29	.5674	1024.1
−122	.604	.01256	1004.3	−77	29.52	.6137	1024.6
−121	.669	.01391	1004.8	−76	31.81	.6613	1025.0
−120	.735	.01528	1005.2	−75	34.37	.7146	1025.5
−119	.805	.01674	1005.7	−74	37.01	.7694	1025.9
−118	.892	.01854	1006.1	−73	39.96	.8308	1026.4
−117	.989	.02056	1006.6	−72	43.04	.8948	1026.8
−116	1.098	.02283	1007.0	−71	46.33	.9632	1027.3
−115	1.208	.02511	1007.5	−70	49.87	1.037	1027.7
−114	1.317	.02738	1007.9	−69	53.59	1.114	1028.2
−113	1.444	.03002	1008.4	−68	57.65	1.199	1028.6
−112	1.575	.03274	1008.8	−67	61.81	1.285	1029.1
−111	1.728	.03593	1009.3	−66	66.41	1.381	1029.5
−110	1.889	.03927	1009.7	−65	71.17	1.480	1030.0
−109	2.087	.04339	1010.2	−64	76.64	1.593	1030.4
−108	2.292	.04765	1010.6	−63	82.28	1.711	1030.9
−107	2.511	.05220	1011.1	−62	88.19	1.833	1031.3
−106	2.742	.05701	1011.5	−61	94.62	1.967	1031.8
−105	2.983	.06202	1012.0	−60	101.4	2.108	1032.2
−104	3.258	.06773	1012.4	−59	108.8	2.262	1032.7
−103	3.543	.07366	1012.9	−58	116.3	2.418	1033.1
−102	3.872	.08050	1013.3	−57	124.8	2.595	1033.6
−101	4.213	.08759	1013.8	−56	133.4	2.773	1034.0
−100	4.607	.09578	1014.2	−55	143.0	2.973	1034.5
−99	5.018	.1043	1014.7	−54	153.0	3.181	1034.9
−98	5.455	.1134	1015.1	−53	163.5	3.399	1035.4
−97	5.946	.1236	1015.6	−52	174.9	3.636	1035.8
−96	6.470	.1345	1016.0	−51	187.0	3.888	1036.3
−95	7.047	.1465	1016.5	−50	199.9	4.156	1036.7
−94	7.638	.1588	1016.9	−49	213.0	4.428	1037.2
−93	8.316	.1729	1017.4	−48	227.9	4.738	1037.6
−92	9.017	.1875	1017.8	−47	243.1	5.054	1038.1
−91	9.806	.2039	1018.3	−46	259.5	5.395	1038.5
−90	10.64	.2212	1018.7	−45	276.7	5.753	1039.0
−89	11.53	.2397	1019.2	−44	295.0	6.133	1039.4
−88	12.51	.2601	1019.6	−43	314.7	6.543	1039.9
−87	13.53	.2813	1020.1	−42	335.3	6.971	1040.3
−86	14.69	.3054	1020.5	−41	357.6	7.435	1040.8

[a]Vapor pressures converted from *International Critical Tables.*

Table 6. Properties of Saturated Water Vapor at Low Temperatures[a] (Con'd.)

Barometer, 29.92 Inches of Mercury

Temperature F	Vapor Pressure In. Hg × 10^{-5}	Weight of Saturated Vapor per Lb Dry Air × 10^{-5}	Btu per Lb of Vapor (32 F Datum)	Temperature F	Vapor Pressure In. Hg × 10^{-5}	Weight of Saturated Vapor per Lb Dry Air × 10^{-5}	Btu per Lb of Vapor (32 F Datum)
−40	380.3	7.907	1041.2	−20	1262.0	26.25	1050.2
−39	405.5	8.431	1041.7	−19	1337.	27.81	1050.7
−38	431.2	8.965	1042.1	−18	1416.	29.45	1051.1
−37	459.2	9.548	1042.6	−17	1496.	31.12	1051.6
−36	488.4	10.16	1043.0	−16	1584.	32.95	1052.0
−35	519.5	10.80	1043.5	−15	1675.	34.84	1052.5
−34	552.4	11.49	1043.9	−14	1772.	36.86	1052.9
−33	586.5	12.20	1044.4	−13	1874.	38.98	1053.4
−32	623.7	12.97	1044.8	−12	1980.	41.19	1053.8
−31	661.8	13.76	1045.3	−11	2093.	43.54	1054.3
−30	701.0	14.58	1045.7	−10	2210.	45.98	1054.7
−29	742.2	15.43	1046.2	−9	2335.	48.58	1055.2
−28	791.2	16.45	1046.6	−8	2463.	51.25	1055.6
−27	841.0	17.49	1047.1	−7	2502.	52.06	1056.1
−26	892.1	18.55	1047.5	−6	2745.	57.12	1056.5
−25	946.4	19.68	1048.0	−5	2898.	60.30	1057.0
−24	1003.	20.86	1048.4	−4	3055.	63.57	1057.4
−23	1064.	22.13	1048.9	−3	3222.	67.05	1057.9
−22	1126.	23.42	1049.3	−2	3397.	70.69	1058.3
−21	1192.	24.79	1049.8	−1	3580.	74.50	1058.8
				0	3773.	78.52	1059.2

[a]Vapor pressures converted from *International Critical Tables.*

W = weight of water vapor mixed with each pound of dry air, pounds.
h'_{fg} = latent heat of vaporization at t', Btu per pound.

Since this definition holds for any mixture of dry air and water vapor, the total heat of a mixture with a relative humidity of 100 per cent and at a temperature equal to the wet-bulb temperature (t') is

$$\Sigma' = c_{p_a}(t' - 0) + W_{t'}\, h'_{fg} \tag{11}$$

By equating Equation 10 to Equation 11, the equation for the adiabatic saturation process, Equation 9a, follows. This demonstrates that the adiabatic saturation process at constant wet-bulb temperature is also a process of constant total heat. In short, the total heat of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. This fact furnishes a convenient means of finding the total heat of an air-vapor mixture in any state.

Example 5. Find the total heat of an air-vapor mixture having a dry-bulb temperature of 85 F and a wet-bulb temperature of 70 F.

Solution. From Table 5, for saturation at the wet-bulb temperature $W_{t'} = 0.01578$, and from Equation 11,

$$\Sigma' = c_{p_a}(70 - 0) + 0.01578\, h'_{fg} = 16.9 + 16.61 = 33.51$$

By considering the temperatures in Table 5 to be wet-bulb readings, the total heat of any air-vapor mixture may be obtained from the last column in the table.

Enthalpy

This total heat of an air-vapor mixture is not exactly equal to the true heat content or enthalpy of the mixture since the heat content of the liquid is not included in Equation 10. With the meaning of heat content in agreement with present practise in other branches of thermodynamics, the true heat content of a mixture of dry air and water vapor (with 0 F as the datum for dry air, and the saturated liquid at 32 F as the datum for the water vapor) is

$$h = c_{p_a} (t - 0) + W h_s = 0.24 (t - 0) + W h_s \tag{12}$$

where

h = the heat content of the mixture, Btu per pound of dry air.
t = the dry-bulb temperature, degrees Fahrenheit.
W = the weight of vapor per pound of dry air, pounds.
h_s = the heat content of the vapor in the mixture, Btu per pound.

The heat content of the water vapor in the mixture may be found in steam charts or tables when the dry-bulb temperature and the partial pressure of the vapor are known. Or, since the heat content of steam at low partial pressures, whether super-heated or saturated, depends only upon temperature, the following empirical equation, derived from Keenan's Steam Tables, may be used:

$$h_s = 1059.2 + 0.45\, t \tag{13}$$

Substituting this value of h_s in Equation 12, the heat content of the mixture is

$$h = 0.24 (t - 0) + W (1059.2 + 0.45\, t) \tag{14}$$

An energy equation can be written that applies, in general, to various air-conditioning processes, and this equation can be used to determine the quantity of heat transferred during such processes. In the most general form, this equation may be explained with the aid of Fig. 1 as follows:

The rectangle may represent any apparatus, *e.g.*, a drier, humidifier, dehumidifier, cooling tower, or the like, by proper choice of the direction of the arrows.

In general, a mixture of air and water vapor, such as atmospheric air, enters the apparatus at 1 and leaves at 3. Water is supplied at some temperature, t_2. For the flow of 1 lb of dry air (with accompanying vapor) through the apparatus, provided there is no appreciable change in the elevation or velocity of the fluids and no mechanical energy delivered to or by the apparatus,

$$h_1 + E_h + (W_3 - W_1)\, h_2 = h_3 + R_c$$

or

$$E_h - R_c = h_3 - h_1 - (W_3 - W_1)\, h_2 \tag{15}$$

where

E_h = the quantity of heat supplied per pound of dry air, Btu.
R_c = the quantity of heat lost externally by heat transfer from the apparatus, Btu per pound of dry air.

W_1 = the weight of water vapor entering, per pound of dry air.
W_3 = the weight of water vapor leaving, per pound of dry air.
h_2 = the heat content of the water supplied at t_2, Btu per pound.
$h_3 - h_1$ = the increase in the heat content of the air-water vapor mixture in passing through the apparatus, Btu per pound of dry air
$= 0.24\ (t_3 - t_1) + W_3\ (1059.2 + 0.45\ t_3) - W_1\ (1059.2 + 0.45t_1)$

The net quantity of heat added to or removed from air-water vapor mixtures in air conditioning work is frequently *approximated* by taking the differences in total heat at exit and entrance.

For example, in Fig. 1, an *approximate* result is

$$E_h - R_c = \Sigma_3 - \Sigma_1 \qquad (16)$$

where

Σ_3 = the total heat of the air-vapor mixture at exit, Btu per pound of dry air.
Σ_1 = the total heat of the air-vapor mixture at entrance, Btu per pound of dry air.

From the definitions of *total heat* and *heat content*, it may be demonstrated that Equation 16 is exactly equivalent to Equation 15, when, and only when, $t'_3 = t'_1 = t_2$; *i.e.*, when the initial and final wet-bulb temperatures and the temperature of the water supplied are equal. The one process that meets these conditions is adiabatic saturation, and either equation will give a result of zero; for other conditions, Equation 16 is approximate but satisfactory for many calculations.

The following problems illustrate the application of these principles:

Example 6. Heating (data from Example 2). Assuming the water to be supplied at 50 F, the net quantity of heat supplied is, from Equation 15,

$$E_h - R_c = 0.24\ (70 - 0) + 0.000547 \times 0.45\ (70 - 0) + 0.005633$$

or

$$1059.2 + 0.45 \times 70 - (50 - 32) = 22.87 \text{ Btu per pound of dry air.}$$

Example 7. Cooling (data from Example 3). If the condensate is removed at 54 F the quantity of heat removed is found from Equation 15, by proper regard to the arrow direction in Fig. 1,

$$E_h + R_c = 0.24\ (84 - 54) + 0.00887 \times 0.45\ (84 - 54) + 0.00358$$

or

$$1059.2 + 0.45 \times 84 - (54 - 32) = 11.17 \text{ Btu per pound of dry air.}$$

Using Table 5, the initial total heat of the air-vapor mixture, since the wet-bulb temperature is 70 F, is 33.51 Btu per pound of dry air.

The final total heat is, from Table 5, since the exit air is saturated, 22.45 Btu per pound. Hence, using Equation 16, the quantity of heat removed is, approximately, (33.51 − 22.45) or 11.06 Btu per pound of dry air. The degree of approximation to the correct result is evident in this example.

PSYCHROMETRIC CHART[4]

The Bulkeley Psychrometric Chart[5], as revised will be found as an insert between pages 18 and 19. It shows graphically the relationships expressed in Equations 9a and 9b. It also gives the grains of moisture per

[4]See A Review of Psychrometric Charts, C. O. Mackey (*Heating and Ventilating*, June, July, 1931)
[5]The Bulkeley Psychrometric Chart was presented to the Society in 1926. (See A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926.) Single copy of the chart can be furnished at a cost of $ .50.

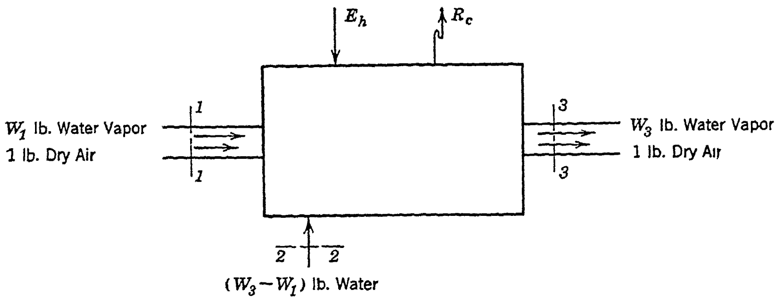

FIG. 1. DIAGRAM ILLUSTRATING ENERGY EQUATION 15

pound of dry air for saturation, the grains of moisture per cubic foot of saturated air, the total heat in Btu per pound of dry air saturated with moisture, and the weight of the dry air in pounds per cubic foot. Fig. 2 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

A is the total heat in Btu contained in the mixture above 0 F, and is to be referred to the column of figures at the left side of the chart. Heat of the liquid is not included.

B is the grains of moisture of water vapor contained in each pound of the saturated mixture and is to be referred to the figures at the left side of the chart.

C is the grains of moisture of water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart which are to be divided by 10.

D is the weight in decimal fractions of a pound, of one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170 and 180 F. The relative density of

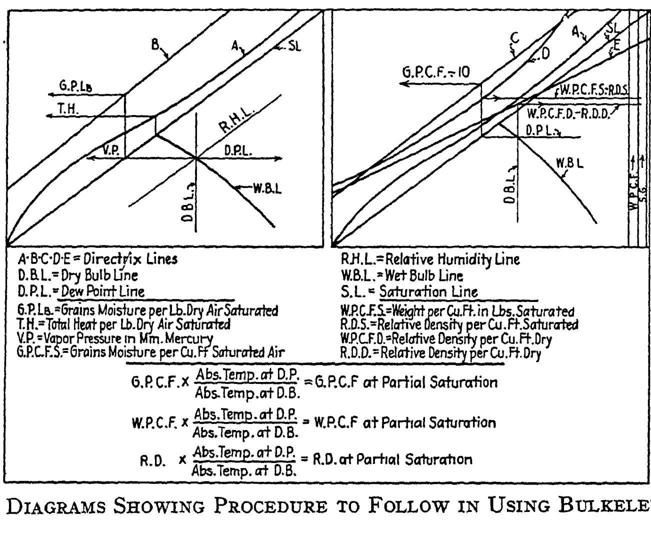

FIG. 2. DIAGRAMS SHOWING PROCEDURE TO FOLLOW IN USING BULKELEY CHART

the mixture is read in a similar manner from the same curve by the column of figures between the vertical dry-bulb temperature lines 180 and 190 F.

E is similar to *D* but is for dry air, devoid of all moisture or water vapor. For convenience, the approximate absolute temperature of 500 F is given at 40 F on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.

METHOD OF USING THE CHART

Example 8. Relative Humidity: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 46 per cent.

Example 9. Dew Point: At the intersection of the 78 F wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70.9 F.

Example 10. Vapor Pressure: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, pass in a horizontal direction to the left of the chart and on the logarithmic scale read the vapor pressure as 19.4 millimeters of mercury. (Divide by 25.4 for inches.)

Example 11. Total Heat Above 0 F in Mixture per Pound of Dry Air Saturated with Moisture: From where the wet-bulb line joins the saturation line, pass in a vertical direction on the 78 F dry-bulb line to its intersection with curve *A* and on the logarithmic scale at the left of the chart read 40.6 Btu per pound of mixture. The use of this curve to obtain the total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of Btu required to heat the mixture and humidify it, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

Example 12. Grains of Moisture per Pound of Mixture: From 70.9 F dew-point temperature on the saturation line, pass vertically to the intersection with curve *B* and on the logarithmic scale at the left read 114 grains of moisture per pound.

Example 13. Grains of Moisture per Cubic Foot of Mixture, Partially Saturated: From 70.9 F dew-point temperature on the saturation line proceed in a vertical direction to curve *C*, and on the logarithmic scale to the left read 83.3 which, divided by 10, gives 8.33 grains. A temperature of 70.9 F is equal to an absolute temperature of 530.9, and 95 F equals 555, absolute temperature. Therefore, $\frac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture.

Example 14. Grains of Moisture per Cubic Foot of Dry Air, Saturated: Starting at the saturation line at the desired temperature, pass in a vertical direction to curve *C* and on the logarithmic scale at the left, read a number which, divided by 10, will give the answer.

Example 15. Weight per Cubic Foot of Dry Air and Relative Density: From the point where, for example, the 70 F vertical dry-bulb line intersects curve *E*, pass to right side and read 0.075 lb; if cubic feet per pound are desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

Example 16. Weight per Cubic Foot of Saturated Air and Relative Density: From the point where, for example, the 70 F vertical line intersects the curve *D*, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70 F.

Example 17. Weight per Cubic Foot and Relative Density of Partially Saturated Air: Air at 50 F and a wet-bulb temperature of 46 F is to be heated to 130 F. The wet- and dry-bulb lines intersect at a dew-point temperature of 42 F. Pass to the left where this dew-point line intersects the saturation line and then pass in a vertical direction to where the 42 F dry-bulb line intersects with curve *D*. Then pass directly to the right and read the weight per cubic foot of saturated air at 42 F as 0.07844 and the relative density as 1.046. The absolute temperature at 42 F is 502, and at 130 F is 590. Therefore, $\frac{502}{590} = 0.851$. The weight of 1 cu ft of air at 50 F dry-bulb and 46 F wet-bulb when heated to 130 F is $0.07844 \times 0.851 = 0.06675$, and the relative density is $1.046 \times 0.851 = 0.89$.

PROPERTIES OF STEAM

Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. This change in state takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The volume of a pound of steam is the *specific volume* which decreases as the pressure increases. The reciprocal of this, or the weight of steam per cubic foot, is the *density*. (See Table 7.)

Steam which is in contact with the water from which it was generated is known as *saturated steam*. If it contains no actual water in the form of mist or priming, it is called *dry saturated steam*. If this be heated and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated*, that is, its temperature will be higher than that of saturated steam at the same pressure.

PROPERTIES OF WATER

Composition of Water. Water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and 16 parts by weight of oxygen.

Density of Water. Water has its greatest density at 39.2 F, and it expands when heated or cooled from this temperature. At 62 F a U. S. gallon of 231 cu in. of water weighs approximately $8\frac{1}{3}$ lb, and a cubic foot of water is equal to 7.48 gal. The specific volume of water depends on the temperature and it is always the reciprocal of its density. (See Table 8.)

Water Pressures. Pressures are often stated in feet or inches of water column. At 62 F, with h equal to the head in feet, the pressure of a column of water is $62.383h$ lb per square foot, or $0.433h$ lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of one pound per square inch at 62 F.

Boiling Point of Water. The boiling point of water varies with the pressure; it is lower at higher altitudes. A change in pressure will always be accompanied by a change in the boiling point, and there will be a corresponding change in the latent heat of evaporation. These values are given in Table 7.

Specific Heat. The specific heat of water, or the amount of heat (Btu) required to raise the temperature of one pound of water one degree Fahrenheit, varies with the temperature, but it is commonly assumed to be unity at all temperatures. Steam tables are based on exact values, however. The specific heat of ice at 32 F is 0.492 Btu per pound. The amount of heat required to raise one pound of water at 32 F through a known temperature interval depends on the average specific heat for the temperature range.

Sensible and Latent Heat. The heat necessary to raise the temperature of one pound of water from 32 F to the boiling point is known as the *heat of the liquid* or *sensible heat*. When more heat is added, the water begins to evaporate and expand at constant temperature until the water is entirely changed into steam. The heat thus added is known as the *latent heat of evaporation*.

Table 7. Properties of Saturated Steam: Pressure Table[a]

		Specific Volume			Total Heat			Entropy			
Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Abs. Press. Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
½″ Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	½″ Hg
¾″ Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	¾″ Hg
1″ Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1″ Hg
1½″ Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	1½″ Hg
2″ Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	2″ Hg
2½″ Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	2½″ Hg
3″ Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3″ Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	96.0
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0

[a]Abstracted from *Steam Tables and Mollier Diagram*, by Prof. J. H. Keenan, 1930 edition, by permission of the publisher, *The American Society of Mechanical Engineers*.

TABLE 7. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume Sat. Liquid	Specific Volume Evap.	Specific Volume Sat. Vapor	Total Heat Sat. Liquid	Total Heat Evap.	Total Heat Sat. Vapor	Entropy Sat. Liquid	Entropy Evap.	Entropy Sat. Vapor	Abs. Press. Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	**100.0**
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	**102.0**
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	**104.0**
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	**106.0**
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	**108.0**
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	**110.0**
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	**112.0**
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	**114.0**
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	**116.0**
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	**118.0**
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	**120.0**
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	**122.0**
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	**124.0**
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	**126.0**
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	**128.0**
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	**130.0**
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	**132.0**
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	**134.0**
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	**136.0**
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	**138.0**
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	**140.0**
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	**142.0**
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	**144.0**
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	**146.0**
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	**148.0**
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	**150.0**
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	**152.0**
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	**154.0**
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	**156.0**
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	**158.0**
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	**160.0**
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	**162.0**
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	**164.0**
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	**166.0**
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	**168.0**
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	**170.0**
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	**172.0**
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	**174.0**
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	**176.0**
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	**178.0**
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	**180.0**
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	**182.0**
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	**184.0**
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	**186.0**
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	**188.0**
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	**190.0**
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	**192.0**
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	**194.0**
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	**196.0**
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	**198.0**
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	**200.0**
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	**205.0**
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	**210.0**
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	**215.0**
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	**220.0**
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	**225.0**
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	**230.0**
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	**235.0**
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	**240.0**
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	**245.0**

TABLE 7. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	**250.0**
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	**260.0**
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	**270.0**
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	**280.0**
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	**290.0**
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	**300.0**
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	**320.0**
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	**340.0**
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	**360.0**
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	**380.0**
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	**400.0**
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	**420.0**
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	**440.0**
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	**460.0**
480.0	462.80	0.0197	0.9456	0.9633	444.9	759.0	1203.9	0.6441	0.8228	1.4670	**480.0**
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	**500.0**
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	**520.0**
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	**540.0**
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	**560.0**
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	**580.0**
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	**600.0**
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	**620.0**
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	**640.0**
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	**660.0**
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	**680.0**
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	**700.0**
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	**720.0**
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	**740.0**
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	**760.0**
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	**780.0**
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	**800.0**
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	**820.0**
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	**840.0**
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	**860.0**
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	**880.0**
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	**900.0**
920.0	534.56	0.0214	0.4635	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	**920.0**
940.0	537.13	0.0215	0.4520	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	**940.0**
960.0	539.66	0.0216	0.4409	0.4625	539.6	651.5	1191.1	0.7410	0.6520	1.3930	**960.0**
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	**980.0**
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	**1000.0**
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	**1050.0**
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	**1100.0**
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	**1150.0**
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	**1200.0**
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	**1250.0**
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	**1300.0**
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	**1350.0**
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	**1400.0**
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	**1450.0**
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	**1500.0**
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	**1600.0**
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	**1700.0**
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	**1800.0**
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	**1900.0**
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	**2000.0**
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	**2200.0**
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	**2400.0**
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	**2600.0**
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	**2800.0**
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	**3000.0**
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	**3200.0**
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	**3226.0**

RATE OF EVAPORATION

In problems of air conditioning and drying, as well as in other industrial applications of evaporation, such as cooling towers, it is desirable to determine the rate of evaporation. There are two distinct cases of evaporation. The *first* case is that in which the source of heat is primarily from the water itself and in which the air temperature may even be raised.

Table 8. Thermal Properties of Water

Temperature Deg F	Sat. Press. Lb per Sq In.	Volume Cu Ft per Lb	Weight Lb per Cu Ft	Specific Heat
32	0.0887	0.01602	62.42	1.0093
40	0.1217	0.01602	62.42	1.0048
50	0.1780	0.01602	62.42	1.0015
60	0.2561	0.01603	62.38	0.9995
70	0.3628	0.01605	62.31	0.9982
80	0.5067	0.01607	62.23	0.9975
90	0.6980	0.01610	62.11	0.9971
100	0.9487	0.01613	62.00	0.9970
110	1.274	0.01616	61.88	0.9971
120	1.692	0.01620	61.73	0.9974
130	2.221	0.01625	61.54	0.9978
140	2.887	0.01629	61.39	0.9984
150	3.716	0.01634	61.20	0.9990
160	4.739	0.01639	61.01	0.9998
170	5.990	0.01645	60.79	1.0007
180	7.510	0.01650	60.61	1.0017
190	9.336	0.01656	60.39	1.0028
200	11.525	0.01663	60.13	1.0039
210	14.123	0.01669	59.92	1.0052
212	14.696	0.01670	59.88	1.0055
220	17.188	0.01676	59.66	1.0068
240	24.97	0.01690	59.17	1.0104
260	35.43	0.01706	58.62	1.0148
280	49.20	0.01723	58.04	1.020
300	67.01	0.01742	57.41	1.026
350	134.62	0.01797	55.65	1.044
400	247.25	0.01865	53.62	1.067
450	422.61	0.0195	51.3	1.095
500	681.09	0.0205	48.8	1.130
550	1045.4	0.0219	45.7	1.200
600	1544.6	0.0241	41.5	1.362
700	3096.4	0.0394	25.4	....

The *second* is that in which the heat for evaporation is obtained entirely from the air itself, in which case the air is cooled and the temperature of the water remains substantially constant at the wet-bulb temperature. Both cases, however, may be reduced to a common basis of calculation. It has been found that the increase in the rate of evaporation is nearly in direct proportion to the increase in the air velocity, and that it is in direct proportion to the difference in vapor pressure between the vapor pressure of the water and the pressure of the vapor in the air.

The general formula covering the experimental data may be expressed as follows:

$$\frac{dw}{dt} = (a + bv)\ (e' - e) \tag{17}$$

where

$\frac{dw}{dt}$ = rate of evaporation.
a = the rate of evaporation in still air.
b = the rate of increase with velocity.
e' = the vapor pressure of the liquid.
e = the vapor pressure in the atmosphere.
v = velocity.

The only difference between case *one* and case *two* is that in case *one* the vapor pressure of the liquid is one of the known or assumed factors, being dependent upon the known temperature of the liquid, while in case *two*, e' is the vapor pressure corresponding to the wet-bulb temperature of the air.

This wet-bulb or evaporation temperature is dependent upon the dry-bulb temperature and the moisture content, or upon the total heat of the air as indicated in the previous paragraph.

The effect of air velocity depends upon whether the flow of air is parallel to the surface or perpendicular to the surface elements. For a flow of air *parallel* to a horizontal surface

$$w = 0.093\left(1 + \frac{v}{230}\right)(e' - e) \quad \text{(approximately)} \tag{18}$$

where

w = pounds evaporated per square foot per hour.
v = velocity of atmosphere over surfaces, feet per minute.
e' = vapor pressure of the water corresponding to its temperature.
e = vapor pressure in the surrounding atmosphere.

For *transverse* flow, as across a tubular surface, the rate of evaporation is nearly doubled.

These relationships are indicated graphically on the chart, Fig. 3.

Since the difference in vapor pressures is substantially proportional to the difference between the wet- and dry-bulb temperatures (*i.e.*, the wet-bulb depression) the rate of evaporation is also, for case *two*, substantially proportionate to the wet-bulb depression.

In case *two*, the rate of sensible heat transfer from the air to the liquid to produce evaporation is substantially the same as the rate of heat transfer with the same type of surface, without moisture being present, but with the same temperature differences. In other words, the rate of heat transfer depends upon the temperature difference only, whether the surface is wet or not. For example, it has been shown that the rate of heat transfer with air flowing across staggered coils (transverse flow) may be represented by the formula:

$$U_t = \frac{1}{0.0447 + \frac{50.66}{v}} \tag{19}$$

where

U_t = heat transfer expressed in Btu per hour per square foot per degree difference in temperature between steam and air, for transverse flow.

At a velocity of 400 fpm, U_t = 5.8; at a velocity of 800 fpm, U_t = 9.3.

Referring to Fig. 3, showing the rate of heat transmission by evaporation for different air velocities, it will be noted that for transverse flow there are 560 Btu per hour per square foot transferred per inch difference of vapor pressure at a velocity of 400 fpm, and 910 Btu per hour per square foot per inch difference in vapor pressure at a velocity of 800 fpm. One inch of vapor pressure difference corresponds approximately to 95 deg difference between the wet- and dry-bulb temperature. Dividing by 95,

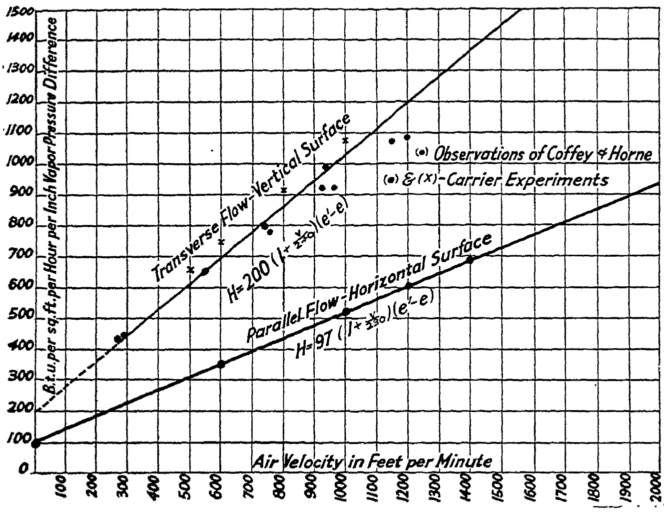

FIG. 3. HEAT TRANSMITTED BY EVAPORATION

the value of 5.9 Btu per square foot per degree difference in temperature is obtained for a velocity of 400 fpm, and 9.55 Btu per square foot for a velocity of 800 fpm.

It will be noted that for these two cases the heat transfer by evaporation per degree difference in temperature corresponds almost exactly with the heat transfer by convection coils. The similarity may be noted by comparing the formula for heat transfer in parallel flow, where

$$U_p = \frac{1}{0.026 + \frac{161}{v}} \tag{20}$$

with the heat transfer by evaporation with parallel flow. The relationship will be seen to be very close in both cases and would indicate that the heat transfer by evaporation is actually brought about by a process of convection.

The difference in form of the two formulae may be due in part to errors in observation at the higher and lower velocities.

In cooling air and condensing out the moisture therefrom the heat transfer is considerably more rapid than when the air is dry and no moisture is condensed. In general the rate of heat transmission on the air side is increased an amount which is proportionate to the latent heat removed as compared with the sensible heat removed. That is, if the latent heat removed was 50 per cent of the sensible heat removed, then the conductivity of the surface in contact with the air would be increased approximately 50 per cent.

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PROBLEMS IN PRACTICE

1 ● Given air at 70 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.00 in. Hg, find the weight of vapor per pound of dry air.

Weight of saturated vapor per pound of dry air = W_t = 0.01578 lb (Table 5). Saturation pressure of the vapor at 70 F = e_t = 0.7386 in. Hg.

From Equation 7,

$$W = \frac{0.01578 \times 0.5\ (29.00 - 0.7386)}{29.00 - (0.5)\ (0.7386)}$$

W = 0.00779 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

Approximate Method:

0.01578 × 0.5 = 0.00789 lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

2 ● Given air with a dry-bulb temperature of 80 F, relative humidity of 55 per cent, and a barometric pressure of 29.92 in. Hg, calculate the weight of a cubic foot of the mixture.

Weight of saturated vapor per cubic foot = 0.001580 lb (Table 5).

$0.001580 \times 0.55 = 0.000869$ lb = weight of vapor per cubic foot at 55 per cent relative humidity.

Pressure of saturated vapor at 80 F = 1.0314 in. Hg.

Pressure of the vapor in the mixture = $1.0314 \times 0.55 = 0.567$ in. Hg.

Pressure of the dry air in the mixture = $29.92 - 0.567 = 29.353$ in. Hg.

Weight of 1 cu ft of dry air at 80 F = $\frac{1}{13.60} = 0.073529$ lb.

Weight of dry air in 1 cu ft of the mixture = $0.073529 \times \frac{29.353}{29.92} = 0.072136$ lb.

$0.072136 + 0.000869 = 0.073005$ lb = weight of 1 cu ft of the mixture.

3 ● Given air with a dry-bulb temperature of 75 F, a relative humidity of 60 per cent, and a barometric pressure of 29.92 in. Hg, calculate the volume of 1 lb of the mixture.

Weight of saturated vapor per cubic foot = 0.001352 lb (Table 5).

$0.001352 \times 0.6 = 0.0008112$ lb = weight of vapor per cubic foot at 60 per cent relative humidity.

Pressure of saturated vapor at 75 F = 0.8744 in. Hg.

Pressure of vapor in the mixture = $0.8744 \times 0.6 = 0.525$ in. Hg.

Pressure of dry air in the mixture = $29.92 - 0.525 = 29.395$ in. Hg.

Volume of 1 lb of dry air at 75 F = 13.48 cu ft.

Volume of 1 lb of dry air in the mixture = $13.48 \times \frac{29.92}{29.395} = 13.72$ cu ft.

Weight of dry air in 1 cu ft of the mixture = $\frac{1}{13.72} = 0.072886$ lb.

$0.072886 + 0.000811 = 0.073697$ lb = weight of 1 cu ft of the mixture.

$\frac{1}{0.073697} = 13.57$ cu ft = volume of 1 lb of the mixture.

Approximate Method:

Volume of 1 lb of saturated air at 75 F = 13.88 cu ft.
Volume of 1 lb of dry air at 75 F = 13.48 cu ft.

Difference in volume = 0.40 cu ft.

Relative humidity = 60 per cent.

$0.40 \times 0.6 = 0.24$ cu ft.

$13.48 + 0.24 = 13.72$ cu ft = volume of 1 lb of the mixture.

The degree of approximation is evident.

4 ● Given saturated air at a temperature of 75 F and a barometric pressure of 29.92 in. Hg, determine the total heat of the mixture per pound of dry air.

From Equation 11 and Table 5,

Cp_a = mean specific heat at constant pressure of dry air = 0.24.

h_{fg} = latent heat of vaporization at the wet-bulb temperature = 1050.1 Btu per lb.

$W_t{'}$ = weight of water vapor mixed with each pound of dry air = 0.01877 lb.

$\Sigma = 0.24\ (75 - 0) + (0.01877)\ (1050.1)$.

$\Sigma = 37.71$ Btu per lb of dry air.

5 ● Given air at 85 F dry-bulb temperature, 75 F wet-bulb temperature, and a barometric pressure of 29.92 in. Hg; determine the total heat of the mixture per pound of dry air.

From Equation 10 and Table 5,

$Cp_a = 0.24$.

$h'_{fg} = 1050.1$ Btu.

Relative humidity = 62.3 per cent (from psychrometric chart).

$W = 0.02634 \times 0.623 = 0.01641$ grains of moisture per lb of dry air.

$\Sigma = 0.24\,(85 - 0) + 0.01641\,[1050.1 + 0.45\,(85 - 75)]$.

$\Sigma = 37.71$ Btu per pound of dry air.

It will be seen from Questions 4 and 5 that the total heat content is a function of the wet-bulb temperature.

6 ● It is desired to maintain a temperature of 80 F and a relative humidity of 50 per cent in a factory where the equipment gives off 6,000 Btu per hour. If the entering air is at 70 F, determine the relative humidity, and the pounds of air required per hour.

Air at 80 F and 50 per cent relative humidity contains 77 grains of moisture per pound.

At 70 F and 77 grains of moisture per pound, the *relative humidity* is 70 per cent.

Total heat above zero in the mixture at 80 F and 50 per cent relative humidity = 31.2 Btu per pound.

Total heat above zero in the mixture at 70 F and 70 per cent relative humidity = 28.8 Btu per pound.

$31.2 - 28.8 = 2.4$ Btu to be removed per pound of air.

6000 Btu = heat given off by equipment per hour.

$$\frac{6000}{2.4} = 2500 \text{ lb of air required per hour.}$$

7 ● From the data given in Question 6, calculate the approximate cubic feet of air required per minute.

Volume of 1 lb of saturated air at 70 F	=	13.69 cu ft (Table 5)
Volume of 1 lb of dry air at 70 F	=	13.35 cu ft.
Difference in volume	=	0.34 cu ft.

Relative humidity = 70 per cent.

$0.34 \times 0.7 = 0.24$ cu ft.

$13.35 + 0.24 = 13.59$ cu ft, volume of 1 lb of mixture at 70 F and 70 per cent relative humidity (approximate).

From Question 6 the air required per hour = 2500 lb.

$$\frac{2500 \times 13.59}{60} = 566.25 \text{ cu ft per minute required.}$$

8 ● Given 1 lb of dry air at 78 F and a barometric pressure of 29.92 in. Hg; calculate the volume. If the temperature is raised to 96 F and the volume remains constant, what will be the new pressure, P_2, in in. Hg?

$PV = WRT$.

R (for air) = 53.34.

$W = 1$ lb.

P = absolute pressure, pounds per square foot.

$$V = \frac{1 \times 53.34 \times (78 + 460)}{29.92 \times 0.491 \times 144}$$

$V = 13.57$ cu ft = volume of 1 lb.

$$\frac{P_2}{P_1} = \frac{T_2}{T_1}; \quad P_2 = \frac{T_2 P_1}{T_1}$$

$$P_2 = \frac{(96 + 460)\,(29.92 \times 0.491 \times 144)}{(78 + 460)\,(0.491 \times 144)}$$

$P_2 = 30.90$ in. Hg.

9 ● Given saturated air at a temperature of 75 F and a barometric pressure of 29.92 in. Hg; determine the heat content of the mixture per pound of dry air, including the heat content of the liquid above 32 F.

From Equation 12,

$$h = 0.24\ (t - 0) + W\ (1059.2 + 0.45t).$$

where

$h_s = 1059.2 + 0.45t$ (Empirical equation derived from Keenan's Steam Tables.)

$t = 75$ F.

$W = 0.01877$ lb of water vapor (Table 5).

$h = 0.24\ (75 - 0) + 0.01877\ (1059.2 + 0.45 \times 75)$.

$h = 38.51$ Btu per pound of dry air.

Chapter 2

VENTILATION AND AIR CONDITIONING STANDARDS

Vitiation of Air, Heat Regulation in Man, Effects of Heat, Effects of Cold, Temperature Changes, Acclimatization, Warmth and Comfort, Effective Temperature, Comfort Chart, Comfort Line, Comfort Zone, Application of Comfort Chart, A.S.H.V.E. Ventilation Standards, Natural and Mechanical Ventilation, Recirculation, Ultra-Violet Radiation and Ionization, Heat and Moisture Losses

VENTILATION is defined in part as "the process of supplying or removing air by natural or mechanical means to or from any space." (See Chapter 41.) The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. More particularly, it includes the simultaneous control of temperature, humidity, movement, and purity of the air. The term is broad enough to embrace whatever other additional factors may be found desirable for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs. Moreover, according to latest researches[1], there is a marked decrease in both positive and negative ions in the air of occupied rooms.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are much too small even under the worst conditions. The amount of carbon dioxide in air is often used in ventilation work as an index of odors of human origin, but

[1]See A.S.H.V.E. research paper entitled Changes in Ionic Content in Occupied Rooms Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

the information it affords rarely justifies the labor involved in making the observation[2]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired and transpired air is odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons alone, whether æsthetic or physiological, are sufficient to warrant a desire for proper air conditions.

A certain part of the dissemination of disease which occurs in confined spaces is caused by the emission of pathogenic bacteria from infected persons. Infections by droplets from coughing and sneezing constitute a limited mode of transmission in the immediate vicinity of the infected person. Experiments have shown that the mouth spray is a coarse rain which settles down quickly. The contamination is local and the problem is considered to be largely one of contact infection rather than air-borne infection.

The primary factors in air conditioning work, in the absence of any specific contaminating source, are temperature, humidity, air movement and body odors. As compared with these physical factors, the chemical factors are, as a general rule, of secondary importance.

HEAT REGULATION IN MAN

The importance of temperature, humidity and air movement arises from the profound influence which these factors exert upon body temperature, comfort and health. Body temperature is a resultant of the balancing action between its heat production and its heat loss. The heat resulting from the combustion of food within the body maintains its temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, conduction and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. In healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to temperature. Nerve courses carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and little understood at the present time. Coördination of these different mechanisms seems to vary greatly with different air conditions.

[2]Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933, p. 324).

With rising air temperatures up to 75 F or 80 F, metabolism, or internal heat production, is decreased[3], probably by an inhibitory action on heat producing organs, especially the adrenal glands, which seem to exert the major influence on basic combustion processes in the body. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, equally good results may be obtained by increasing the air movement, and hence the heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted, and the blood supply to the skin is curtailed by vasomotor shifts to the internal organs in order to conserve body heat.

EFFECTS OF HEAT

Although the human organism is capable of adapting itself to variations in environmental conditions, its ability to maintain heat equilibrium is limited. The heat regulating center fails, for instance, if the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 1.) In extreme conditions, the metabolic rate is markedly increased owing to the excessive rise in body temperature[4], and a vicious cycle results which may eventually lead to serious physiologic damage.

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the most deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is believed that the feeling of lassitude and discomfort experienced is due to the anæmic condition of the brain.

[3]Heat and Moisture Losses From the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245).

[4]Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (*The American Journal of Physiology*, Vol. 88, No. 3, April, 1929).

TABLE 1. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp.	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hour)	Increase in Pulse Rate (Beats per Min per Hour)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt. by Perspiration (Lb per Hr)
60	....	....	....	....	225,000	0.0	6	0.5
70	....	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

The stomach loses some of its power to act upon the food, owing to a diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[5]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather. The victim may lose appetite and suffer from indigestion, headache and general enervation, which may eventually lead to a premature old age.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss[6] recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride to a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show clearly that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

EFFECTS OF COLD

The action of cold on human beings is not well known. Cold affects the human organism in two ways: (1) through its action on the body as a whole, and (2) through its action on the mucous membranes of the upper respiratory tract. Little exact information is available on the latter.

On exposure to cold, the loss of heat is increased considerably and only

[5]Influence of Effective Temperature upon Bactericidal Action of Gasto-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.* Vol. 24, 1927, p. 832).

[6]Some Effects of High Air Temperatures upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284).

within certain limits is compensation possible by increased heat production and decreased peripheral circulation. The rectal temperature often rises upon exposure to cold but the pulse rate and skin temperature fall. The blood pressure increases, owing to constriction in the peripheral vessels and to thickening of the blood. The subcutaneous tissues and muscles form reservoirs for storing the water which leaves the blood. In extremely cold atmospheres compensation becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

Cannon showed that excessive loss of heat is associated with increased activity of the adrenal medulla[7]. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast[8] found a degeneration of thyroid and adrenal glands upon exposure to cold.

Effects of Temperature Changes

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced in passing from a warm room to the cold air out of doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils and throat, which is accompanied by a fall in the temperature of the tissues. On rewarming, the palate and throat do not always regain their normal temperature and blood supply. This anæmic condition favors bacterial activity and it is believed to play a part in the inception of the common cold and other respiratory diseases. It is believed that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England[9] found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and tin-plate millmen, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries[10]. According to these data the highest

[7]Studies on the Condition of Activity of Endocrine Glands, by W. B. Cannon, A. Guerido, S. W. Britton and E. M. Bright (*American Journal of Physiology*, Vol. 79, 1926, p. 466).

[8]Studies in Exhaustion Due to Lack of Sleep, by T. H. Bast, J. S. Supernaw, B. Lieberman and J. Munro (*American Journal of Physiology*, Vol. 85, 1928, p. 135).

[9]Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (*Industrial Fatigue Research Board*, Report No. 5, 1920, London).

[10]Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin*, Metropolitan Life Insurance Company, 1928).

pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of the present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to a climate averaging between 60 and 80 F is a very primitive trait[11].

Within these limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat have a lower rate of heat production than do those who grow up in cooler environments. This causes them to stand chilling poorly as they are unable to quickly increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adrenal activity has slowed down. Within a couple of years, however, they find themselves standing the heat much better and disliking cold. They become acclimated by a definite change in the combustion level within the body[12].

In certain individuals the psychologic factor is more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed regardless of the quality of the air. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

WARMTH AND COMFORT

The temperature, humidity, and motion of the air, and the radiation between a person and surrounding hot or cold surfaces, taken together, determine his feeling of warmth and influence his elimination of body heat. In other words, the temperature sensations of the human body depend not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by

[11]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1924.

[12]Air Conditioning it its Relation to Human Welfare, by C. A. Mills, M.D. (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, April, 1934).

a wet-bulb thermometer. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler.

On the other hand, in cold environments an increase in humidity produces a cooler sensation. The dividing line at which humidity has no effect upon comfort varies with the air velocity and is about 46 F (dry-bulb) for still air and about 51, 56 and 59 F for air velocities of 100, 300 and 500 fpm, respectively.

Thermo-Equivalent Conditions

Combinations of temperature, humidity and air movement which produce the same feeling of warmth are called thermo-equivalent conditions. A series of tests[13, 14, 15] has been carried out in the psychrometric rooms of the A.S.H.V.E. Research Laboratory, Pittsburgh, in order to determine the equivalent conditions met with in general air conditioning work. These show that this newly-developed scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological *effects* on the body induced by heat and cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an index of warmth or cold. It is not in itself an index of comfort, as it is often assumed to be, nor are the effective temperature lines necessarily lines of equal comfort. This is true because, in determining this index, the subjects compared not the relative comfort, but rather the relative warmth or cold of various air conditions. Moist air at a comparatively low temperature, and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be different, although the effective temperature is the same.

Under extreme humidity conditions there seems to be a difference between sensations of absolute comfort and of the proper degree of warmth. In other words, human beings are not necessarily comfortable when the air is neither *too warm* nor *too cold*. Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discomfort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture. According to the comfort experiments first conducted at the A.S.H.V.E. Laboratory[16] in the U. S. Bureau of Mines, Pittsburgh, and later studies at the Harvard School of Public Health[17] in Boston, effective temperature appears to be a fair index of comfort also, particularly within a humidity range of 30 to 60 per cent, approximately.

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[13]Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yaglogiou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

[14]Cooling Effect on Human Beings by Various Air Velocities, by F. C. Houghten and C. P. Yaglogiou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193).

[15]Effective Temperature with Clothing, by C. P. Yaglogiou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89).

[16]Determination of the Comfort Zone With Further Verification of Effective Temperatures Within This Zone, by F. C. Houghten and C. P. Yaglogiou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

[17]The Summer Comfort Zone; Climate and Clothing, by C. P. Yaglou and Philip Drinker (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

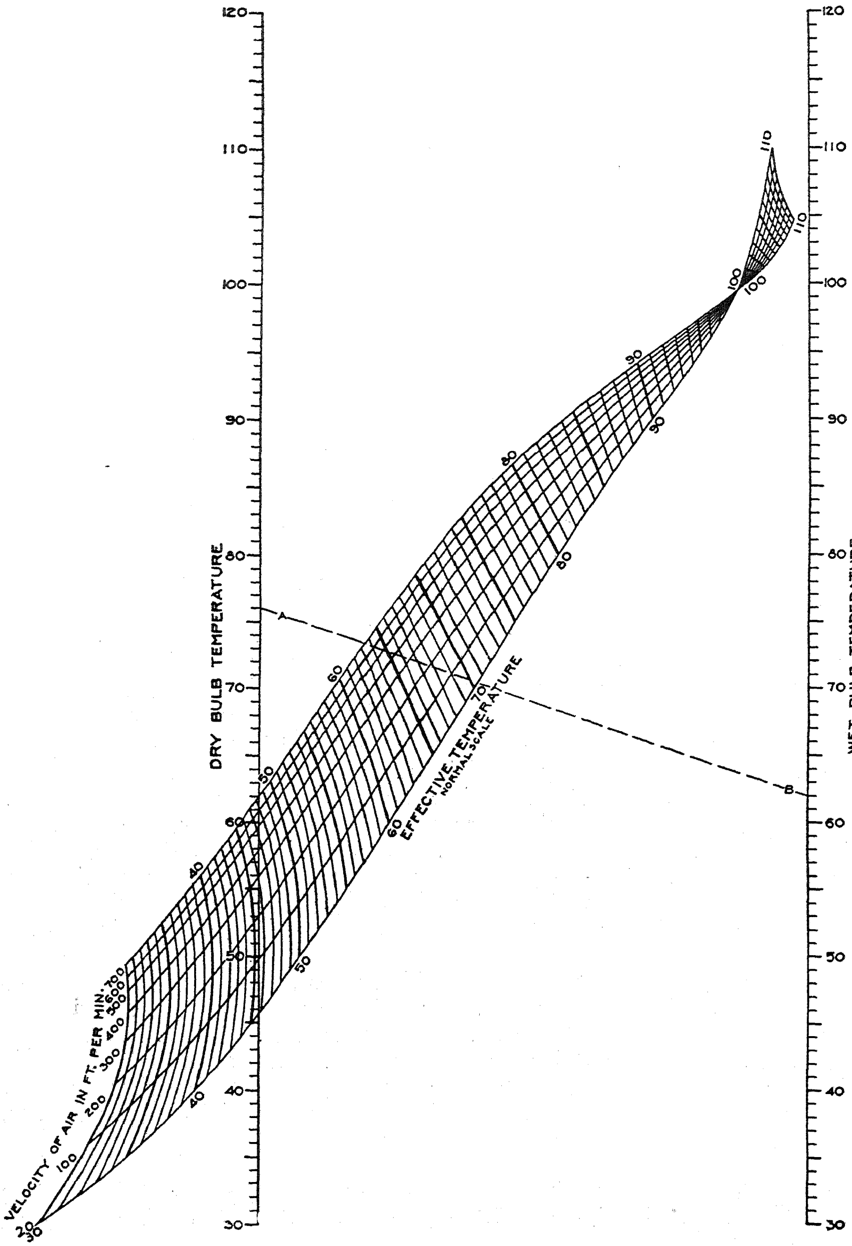

FIG. 1. THERMOMETRIC OR EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.*, warm air, direct steam or hot water radiators, plenum systems.

Definition of Effective Temperature

Briefly, *effective temperature* may be defined as an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is not a true temperature of the air but an index which combines temperature, humidity and air motion in a single value. The numerical value of the effective temperature index for any given air condition is fixed by the temperature of saturated air which, at a velocity or turbulence of 15 to 25 fpm, induces a sensation of warmth or cold like that of the given condition. Thus, any air condition has an effective temperature of 65 deg when it induces a sensation of warmth like that experienced in practically still air at 65 F saturated with moisture.

In all reports of the A.S.H.V.E. Research Laboratory, the term *still air* signifies the minimum air movement it was possible to obtain in the Laboratory's psychrometric chamber. Actually, the air motion was between 15 and 25 fpm in all experiments, without qualification, as measured by the Kata thermometer. This was not a linear movement of air but it represented the turbulence or eddy currents produced by the air change. Even in tightly sealed rooms, the natural air movement is not likely to fall below 10 fpm so long as there is a temperature or pressure difference between the air inside and that outside the room.

Fig. 1 shows the results obtained at the A.S.H.V.E. Research Laboratory in a single chart, the so-called thermometric chart. The equivalent conditions or effective temperature lines are shown by the short cross-lines. The difference between the effective temperature for still air and for moving air, of any velocity, represents the cooling resulting from that air velocity. This thermometric chart applies to average normal and healthy persons adapted to American living and working conditions. It is limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, central fan and direct hot water and steam heating systems) in which the difference between the air and wall surface temperatures may not be great. The chart does not apply to rooms heated by radiant methods such as the British panel system, open coal fires, and the like. It will probably not apply with adequate accuracy to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different from those of the subjects employed in experiments at the Research Laboratory.

If an occupant of a room loses heat by radiation to large wall or glass surfaces at lower temperatures, the air within the room must be maintained at a higher temperature to compensate for this effect in order to give the same feeling of warmth. The results of a recent study[18] by the A.S.H.V.E. Laboratory, shown in Fig. 2, indicate that in poorly insulated buildings this effect may become of considerable importance. Thus an occupant of a room having inside wall surface temperatures of 55 F on three sides will require an air temperature of 74 F to have the same feeling of warmth he would experience in a warm-wall room with air at 70 F. A wall consisting of 8-in. brick and plaster, with 16 F outside air tempera-

[18]Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1933, p. 53).

ture and 70 F inside air temperature, will have an inside surface temperature of 55 F. The reverse effect will be experienced by occupants of rooms having extensive high-temperature surfaces in them. In such cases, a lower air temperature is required to compensate for heat radiated to the occupant.

The effective temperature index for persons doing medium or heavy muscular work, in still air, has also been determined at the A.S.H.V.E. Research Laboratory[19].

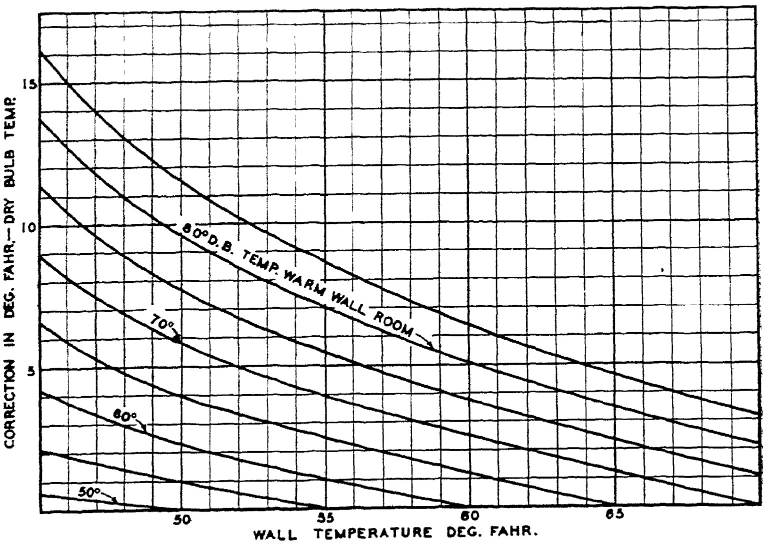

Fig. 2. Correction to Various Dry-Bulb Temperatures in a Warm-Wall Room for the Same Feeling of Warmth in Rooms Having Three Cold Walls. Temperatures Indicated by Shielded Thermometers 30 in. Above the Floor

OPTIMUM AIR CONDITIONS

No single comfort standard can be laid down which would meet every need. There is an inherent individual variation in the sensation of warmth or comfort felt by persons when exposed to an identical atmospheric condition. The state of health, age, sex, clothing, activity, and the degree of acquired adaptation seem to be the important factors affecting the comfort standards.

Since the prolonged effects of temperature, humidity and air movement on health are not known to the same extent as their effects on comfort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, the two do not differ greatly since this is in accordance with the efficient operation of the heat regulating mechanism of the body. This belief is strengthened by

[19]Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. Transactions, Vol. 32, 1926).

results of studies on premature infants over a four-year period[20]. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhœa and mortality was decreased, gains in body weight increased and infections were reduced to a minimum.

Comfort Chart; Comfort Line; Comfort Zone

Fig. 3 shows a comfort chart, developed at the A.S.H.V.E. Laboratory, on which the average and extreme comfort zones have been superimposed. The *extreme comfort zone* includes air conditions in which one or more of the experimental subjects were comfortable. The *average comfort zone* includes those air conditions in which the majority of the subjects (50 per cent or more) were comfortable. That particular effective temperature at which the maximum number of subjects was comfortable was called the *comfort line.*

The *average winter comfort zone* as determined at the A.S.H.V.E. Laboratory ranges from 63 deg to 71 deg ET (effective temperature). In winter while at rest, a large percentage of persons normally clothed were found to be comfortable at 66 deg ET and this temperature has been accepted by a committee of the Society[21] as the *winter comfort line* or *optimum effective temperature.*

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal equilibrium with its environment with the least conscious sensation to the individual, or with the minimum phsyiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity, but also the condition of the surrounding objects and wall surfaces. The comfort zone tests were made in rooms with wall surface temperatures approximately the same as the room dry-bulb temperature. For walls of large area having unusually high or low surface temperatures, however, a somewhat lower or higher range of effective temperature is required to compensate for the increased gain or loss of heat to or from the body by radiation[22].

The *average summer comfort zone* for exposures of 3 hours or more ranges from about 66 deg to 75 deg ET, based on studies made at the Harvard School of Public Health[17]. The probable optimum effective temperature (for exposures of 3 hours or more) is 71 deg. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

The best effective temperature (for exposures lasting 3 hours or more) was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approxi-

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[20]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[21]How to Use the Effective Temperature Index and Comfort Charts (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

[22]Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1933, p. 53).

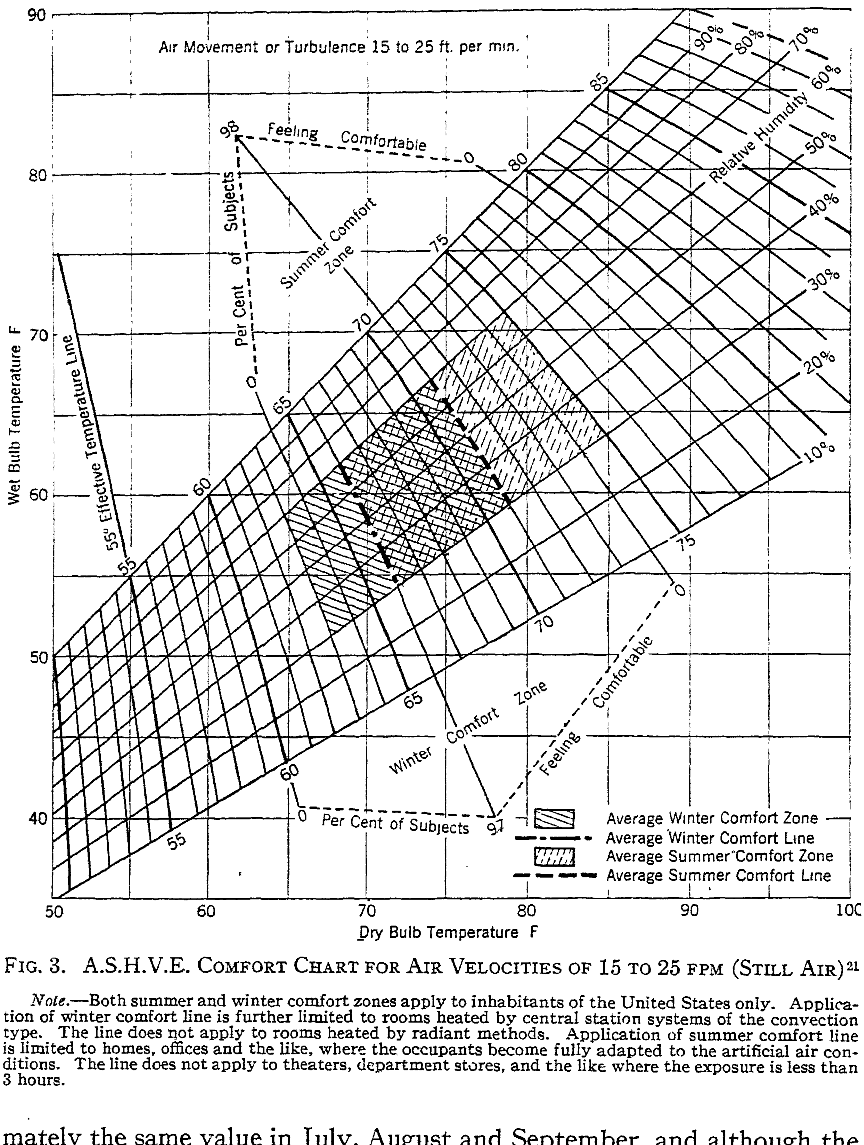

FIG. 3. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)[21]

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

mately the same value in July, August and September, and although the average monthly temperature did not vary much, the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Young men as a general rule prefer conditions in the cool region of the comfort zone, and women and older people in the warm region of the

comfort zone. Crowding the experimental chamber lowered the optimum effective temperature from 70.8 deg when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft, to 69.4 deg when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, a relative humidity of 70 per cent was found to be *somewhat humid* in winter, by about half of the subjects who were stripped to the waist, even when the dry-bulb temperature was 70 F or less. In summer, a relative humidity of 30 per cent was pronounced as *a little too dry* by about a third of the subjects wearing warm-weather clothing. So long as the temperature was kept within proper limits, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 and 60 per cent. This is in accord with studies by Howell[23], Miura[24] and others.

Dry air produces an excessive loss of moisture from the skin and respiratory tract. Owing to the cooling effect of evaporation, higher temperatures are necessary, and this condition leads to discomfort and lassitude. Moist air, on the other hand, interferes with the normal evaporation of moisture from the skin, and again may cause a feeling of oppression and lassitude, especially when the temperature is also high.

Just what the optimum range of humidity is, is a matter of conjecture. There seems to exist a general opinion, supported by some experimental and statistical data, that warm, dry air is less pleasant than air of a moderate humidity, and that it dries up the mucous membranes in such a way as to increase susceptibility to colds and other respiratory disorders[25, 26, 27].

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[28], and according to Huntington[29], this seems to be the case for adults also. All of these studies indicate that the optimum humidity must always be considered in combination with temperature.

Until more exact information is secured, it would be desirable to restrict the comfort zones to the range of relative humidity employed in the comfort zone experiments, namely, 30 to 70 per cent. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort, so long as extremely low humidities are avoided. From the standpoint of health, however, the consensus seems to favor a relative humidity between

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[23]Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).

[24]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

[25]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al (*Journal Experimental Medicine*, 1921, Vol. 34, p. 11).

[26]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A. Goldman, et al and Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, Vol. 29, p. 151).

[27]The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al (Am. Otol., Rhinol., and Laryngol., 1921).

[28]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[29]Weather and Health, by Ellsworth Huntington (*Bulletin* of the National Research Council No. 75. The National Academy of Science, Washington, D. C., 1930).

40 and 60 per cent. In mild weather such comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken to properly insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. Information on the prevention of condensation on building surfaces is given in Chapter 7. The principles underlying humidity requirements and limitations are discussed more fully elsewhere[30].

The comfort chart (Fig. 3) applies to adults between 20 and 70 years of age living in the northeastern parts of the United States. For prematurely born infants, the optimum temperature varies from 100 F to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 F to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 F to 68 F temperature with a moderate humidity (not specified) and a moderate but not excessive amount of air movement (not specified)[31].

Satisfactory comfort conditions are found to vary from 40 deg to 70 deg ET, depending upon the rate of work and amount of clothing worn. The effective temperatures giving maximum comfort for persons working have been determined by the A.S.H.V.E. Research Laboratory[32] for a rate of work which is considered hard labor. For this degree of work, 50 per cent were fairly comfortable for temperatures ranging from 46 to 64 deg ET, while the greatest percentage found maximum comfort at 53 deg ET. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

APPLICATION OF COMFORT CHART

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, or to rooms with excessive glass area or rooms with poorly insulated or cold walls, and it has not been advocated officially for use in foreign countries where the climate, heating methods, and general living conditions are materially different from those in the United States, although several foreign workers have attempted to show that it cannot be so applied. Even in the warm south and southwestern

[30]Humidification for Residences, by A. P. Kratz (*University of Illinois Engineering Experiment Station Bulletin* No. 230, July 28, 1931).

[31]Ventilation, Report of the New York State Commission on Ventilation, 1923.

[32]A.S.H.V.E. research paper entitled Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures are necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants[22] in close proximity. In rooms in which the average wall surface temperature is considerably below the air temperature, higher air temperatures are necessary. The reverse holds true in radiant or panel heating methods. (See Chapter 38.)

The sensation of comfort, in so far as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm*. These individual differences among the minority should be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

Strictly speaking, the only authoritative comfort zone on which accurate data are available, is that for 15 to 25 fpm air movement or turbulance (often referred to as still air). In the past, the winter comfort zone has often been superimposed on the thermometric chart or on effective temperature charts for various air velocities, on the assumption that air conditions of equal warmth are approximately equally comfortable. This may hold in hot industries where the workers are adapted to high temperatures and strong air currents, but it does not apply to sedentary conditions. To ascertain approximately whether a given industrial condition is reasonably comfortable, it would be necessary first to compute the effective temperature from the thermometric chart (Fig. 1) and then to refer this effective temperature to the comfort chart (Fig. 3), or to refer directly to a chart or table for the proper air velocity.

The summer comfort line (71 deg ET) is applicable to the same geographic area as the winter comfort line. It is further restricted to cases in which the human body has reached thermal equilibrium with its environment. As a general rule this takes place after 1½ to 3 hours' exposure. When a person from outdoors enters a room cooled to 71 deg ET on a hot day (95 F or over) an intense chill is likely to be experienced which is unpleasant. However, after remaining in the room for about 2 hours, this fundamental optimum condition will prove satisfactory to the average person. The summer comfort zone, as well as the comfort line, makes proper allowance for these adaptive changes in the body, and thus applies to homes, offices, schools and other similar places where persons of sedentary occupations spend from 3 to 8 or more hours daily.

In artificially cooled theaters, department stores, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is not to reduce the temperature to the optimum degree, but to maintain therein a temperature which is temporarily comfortable to the patrons who thus avoid sensations of chill and intense heat on entering and leaving the building. The relative humidity should be low enough (about 50 per cent or less) to give a sense of comfort without chill and to induce a rate of evaporation which will keep clothing and skin dry. For exposures less than 3 hours, desirable indoor conditions in summer corresponding to various outdoor temperatures are given in Table 2.

It should be kept in mind that southern people, with their more sluggish heat production and lack of adaptability, will demand a comfort zone several degrees higher than those given here for the more active people of northern climates. Instead of the summer comfort line standing at 71 deg as here given, it was found to be much higher for foreigners in Shanghai where climatic conditions are similar to those of our gulf states. This difference in basic metabolic level of people forms a very real problem for air conditioning engineers, which they must recognize in their efforts to give proper conditions of comfort. Cooling of theaters, resturants, and other public buildings in southern climates cannot be based on northern standards without considerable modification.

Table 2. Desirable Indoor Air Conditions in Summer Corresponding to Outdoor Temperatures

Applicable to Exposures Less Than 3 Hours

Outdoor Temperature (Deg Fahr)	Indoor Air Conditions with Dew Point Constant at 57 F		
Dry-Bulb	Dry-Bulb	Wet-Bulb	Effective Temp
95	80.0	65.0	73
90	78.0	64.5	72
85	76.5	64.0	71
80	75.0	63.5	70
75	73.5	63.0	69
70	72.0	62.5	68

A.S.H.V.E. VENTILATION STANDARDS[33]

It is the intent of the Committee in presenting this report to confine itself to a statement of those requirements which, based on present day knowledge, will provide adequate ventilation for spaces intended for human occupancy. The following standards shall apply to all spaces occupied by human beings in all buildings for which ventilation regulations are to be established.

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[33]Report of A.S.H.V.E. Committee on Ventilation Standards consisting of W. H. Driscoll, *Chairman*, J. J. Aeberly, F. Paul Anderson, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard, presented at the Semi-Annual Meeting of the Society, Milwaukee, Wis., June, 1932, and adopted by the Society in August, 1932.

SECTION I—AIR TEMPERATURE AND HUMIDITY

The *temperature* and *humidity* of the air in such occupied spaces, and in which the only source of contamination is the occupant, shall be maintained at all times during occupancy at an *Effective Temperature*, as hereinafter stated.

The relative humidity shall be not less than 30 per cent, nor more than 60 per cent in any case. The Effective Temperature shall range between 64 deg and 69 deg when heating or humidification is required, and between 69 deg and 73 deg when cooling or dehumidification is required.

These Effective Temperatures shall be maintained at a level of 36 in. above the floor. (See Appendix, Tables A and B).

SECTION II—AIR QUALITY

The air in such occupied spaces shall at all times be free from toxic, unhealthful or disagreeable gases and fumes and shall be relatively free from odors and dust.

In every space coming within the provisions of these requirements and in which the quality of the air is below the standards prescribed by good medical and engineering practices, due to toxic substances, bacteria, dust, excessive temperature, excessive humidity, objectionable odors, or other similar causes, means for ventilating shall be provided so that the quality of the air shall be raised to these standards.

SECTION III—AIR MOTION

The air in such occupied spaces shall at all times be in constant motion sufficient to maintain a reasonable uniformity of temperature and humidity, but not such as to cause objectionable drafts in any occupied portion of such spaces.

The air motion in such occupied spaces, and in which the only source of contamination is the occupant, shall have a velocity of not more than 50 feet per minute, measured at a height of 36 in. above the floor.

SECTION IV—AIR DISTRIBUTION

The air in all rooms and enclosed spaces shall, under the provisions of these requirements, be distributed with reasonable uniformity, and the variation in the carbon dioxide content of the air shall be taken as a measure of such distribution.

The air in a space ventilated in accordance with these requirements, and in which the only source of contamination is the occupant, shall be distributed and circulated so that the variation in the concentration of carbon dioxide, when measured at a height of 36 in. above the floor, shall not exceed one part in 10,000.

SECTION V—AIR QUANTITY

The quantity of air used to ventilate the given space during occupancy shall always be sufficient to maintain the standards of air temperature, air quality, air motion and air distribution as herein required. Not less than 10 cubic feet per minute per occupant of the total air circulated to meet these requirements shall be taken from an outdoor source.

APPENDIX

Definitions

For the purposes of these standards the terms used shall be defined as follows:—

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*).

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Dry-Bulb Temperature: The temperature of the air which is indicated by any type of thermometer which is not affected by the water vapor content or relative humidity of the air.

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, such as are blown on a windy day, the average diameter of which is approximately 0.01 centimeter, may be called dust.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite

index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Humidity: The water vapor (either saturated or superheated steam) occupying any space, which may or may not contain other vapors and gases at the same time.

Relative Humidity: A ratio, although usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space, which depends merely on the temperature and partial pressure of the vapor.

Spaces in Which the Only Source of Contamination Is the Occupant: Spaces in which the atmospheric contamination results entirely from the respiratory processes of the occupant, including heat, moisture, and odors given off by the body. No manufacturing or industrial processes or other sources of atmospheric contamination, including heat and moisture, than people are considered under this title.

TABLE A. EFFECTIVE TEMPERATURES RANGING FROM 64 DEG TO 69 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE[a]

(For use when heating or humidification is required)

Dry-Bulb Temperatures (Deg Fahr)	Relative Humidities (Per Cent)						
	30	35	40	45	50	55	60
	Effective Temperatures (Degrees)						
67						64.0	64.3
68			64.0	64.2	64.5	64.8	65.1
69	64.1	64.4	64.8	65.1	65.4	65.7	66.0
70	64.8	65.1	65.4	65.8	66.2	66.5	66.8
71	65.5	65.8	66.2	66.6	67.0	67.3	67.7
72	66.2	66.5	66.9	67.3	67.7	68.1	68.5
73	67.0	67.3	67.7	68.1	68.5	68.9	
74	67.7	68.0	68.4	68.8			
75	68.4	68.7					
76	69.0						

[a]See Fig. 3.

TABLE B. EFFECTIVE TEMPERATURES RANGING FROM 69 DEG TO 73 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE[a-b]

(For use when cooling or dehumidification is required)

Dry-Bulb Temperatures (Deg Fahr)	Relative Humidities (Per Cent)						
	30	35	40	45	50	55	60
	Effective Temperatures (Degrees)						
73							69.3
74					69.3	69.7	70.1
75			69.1	69.5	70.0	71.5	71.0
76	69.0	69.4	69.9	70.5	70.8	71.3	71.8
77	69.7	70.2	70.7	71.2	71.6	72.1	72.6
78	70.4	70.9	71.4	71.9	72.4	73.0	
79	71.1	71.6	72.2	72.6			
80	71.8	72.4	72.9				
81	72.5						

[a]See Fig. 3.

[b]This table applies primarily to cases in which the human body has reached equilibrium with the surrounding air. A higher plane of summer effective temperatures is required in places of public assembly where the period of occupancy is short, than is required for offices and industrial plants where the period of occupancy is of longer duration. When the period of occupancy is two hours or less, the dry-bulb temperature shall be 72 F plus one-third of the difference between the outside dry-bulb temperature and 70 F, and the relative humidity shall not exceed 60 per cent. (See also Table 2.)

FACTORS INFLUENCING APPLICATIONS

The conditions and limitations outlined under the heading Application of Comfort Chart should be noted in applying the temperatures and relative humidities specified in Tables *A* and *B* of the preceding A.S.H.V.E. Ventilation Standards.

Air Quality

In occupied spaces in which the vitiation is entirely of human origin, the chemical composition of the air, the dust, and bacteria content may be dismissed from consideration so that the problem consists in maintaining a suitable temperature with a moderate humidity, and in keeping the atmosphere free from objectionable odors. Such unpleasant odors, human or otherwise, can be easily detected by persons entering the room from clean, odorless air. A further discussion of air quality will be found in Chapters 15 and 16.

Air Motion

As a result of studies by Baetjer[34] and work carried on by the A.S.H.V.E. Research Laboratory, it is now recognized that the importance of air motion in air conditioning ranks only second to temperature. Air in an occupied space having all the other essential qualities but lacking in air motion feels stagnant, stuffy, and depressing, because the vitiated air next to the body is not replaced by the surrounding air possessing the satisfactory qualities. Hence, air motion is absolutely essential that an occupant may realize the other desired qualities of the atmosphere. Possible limits in variation in air motion may range from 5 fpm to 50 fpm, as measured by the Kata thermometer. (See Chapter 40.) However, satisfactory results are more likely to be insured by air velocities ranging from 15 to 30 fpm. The limit of 5 fpm may be taken as the minimum during the heating season, and 50 fpm as the maximum for the cooling season.

Air Distribution

Variation in concentration of carbon dioxide in different parts of an occupied room has been used as a measure of satisfactory distribution of the outside or conditioned air supply. For satisfactory air distribution, the carbon dioxide concentration at the 36-in. level should not vary by more than one part in 10,000 parts of air. Recent work[2] by the A.S.H.V.E. Research Laboratory demonstrates that variations in dry-bulb temperature, wet-bulb temperature, or moisture content of the air are equally good indices of air distribution. This work also indicates that the presence of satisfactory air motion within the room (15 to 30 fpm as measured by the Kata thermometer) insures satisfactory distribution. Because of the laborious and exacting technique involved in making carbon dioxide determinations, it is recommended that satisfactory distribution can be amply insured by the presence of such air velocities in all parts of the room together with dry-bulb temperature variations of not to exceed 3 deg at the 36-in. level.

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[34]Threshold Air Currents in Ventilation (*American Journal of Hygiene*, Vol. IV, No. 6, p. 650, 1924).

Air Quantity

The quantity of air to be circulated through an occupied space, whether by natural or mechanical means, or whether the air is conditioned or not, must in all cases be sufficient to maintain the required standards of air temperature, quality, motion and distribution. The factors which determine air quantity include the type and nature of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

The quantity of air supplied to a room by an air conditioning or ventilating system serves two purposes: *First*, the supply of sufficient outside air for the needs of the occupants; and *second*, the setting up of circulation or air motion within the room. Until recently it was considered that 30 cfm were necessary in any occupied space, particularly in a classroom. It has since been demonstrated that 10 cfm of outside air per person is frequently sufficient to remove body heat, insure against body odors, and provide the chemical needs of respiration. However, it is found that a greater volume should be circulated in the average room in order to provide the required air motion. It is now customary to supply the minimum amount of outside or conditioned air required for removing heat and odors, and to recirculate the additional volume.

In offices and small rooms where the occupants smoke, from 6 to 7 cfm of outside air per occupant will be necessary to eliminate the nuisance effects of the smoke; this quantity of air, however, may be a part of that necessary for other ventilation requirements. Restaurants which permit smoking, because of the exposed food and the necessity that restaurant air seem very clean, need from 10 to 12 cfm of outside air per occupant to care for the smoke condition. This air, likewise, need not be in addition to that required for other ventilation purposes.

Temperature Rise

The total quantity of air introduced is governed largely by the needs for controlling temperature and humidity when either heating or cooling is required. As a rule, the introduction and distribution of warm air into an occupied space does not present as many difficulties as does the introduction of cold air. The former is determined from the amount of heat to be given up to the space, and the latter is determined from the amount of heat to be removed from the space, using a temperature rise that will produce uniform distribution without the production of disagreeable drafts.

Fig. 4 shows the changes in carbon dioxide concentration and moisture content resulting from occupation, in the atmosphere of a room supplied with various volumes of outside air. Data are given for an adult, 5 ft 8 in. in height weighing 150 pounds and having a body surface area of 19.5 sq ft, and for a child, 12 years of age, 4 ft 7 in. in height, weighing 76.6 pounds and having a body surface area of 12.6 sq ft. It is a recognized fact that the dissipation of heat and moisture to the atmosphere, the addition of carbon dioxide, and all metabolic changes take place in proportion to the surface area of the individual. Hence, data for persons of other sizes may be obtained by interpolating among the curves given. The rate of sensible heat production is given in Fig. 7. Fig. 4 also gives

the temperature of the incoming air necessary to maintain a room temperature of either 70 or 80 F as indicated, assuming that there is no heat gain or loss to the room by transmission through the walls, solar radiation or other sources.

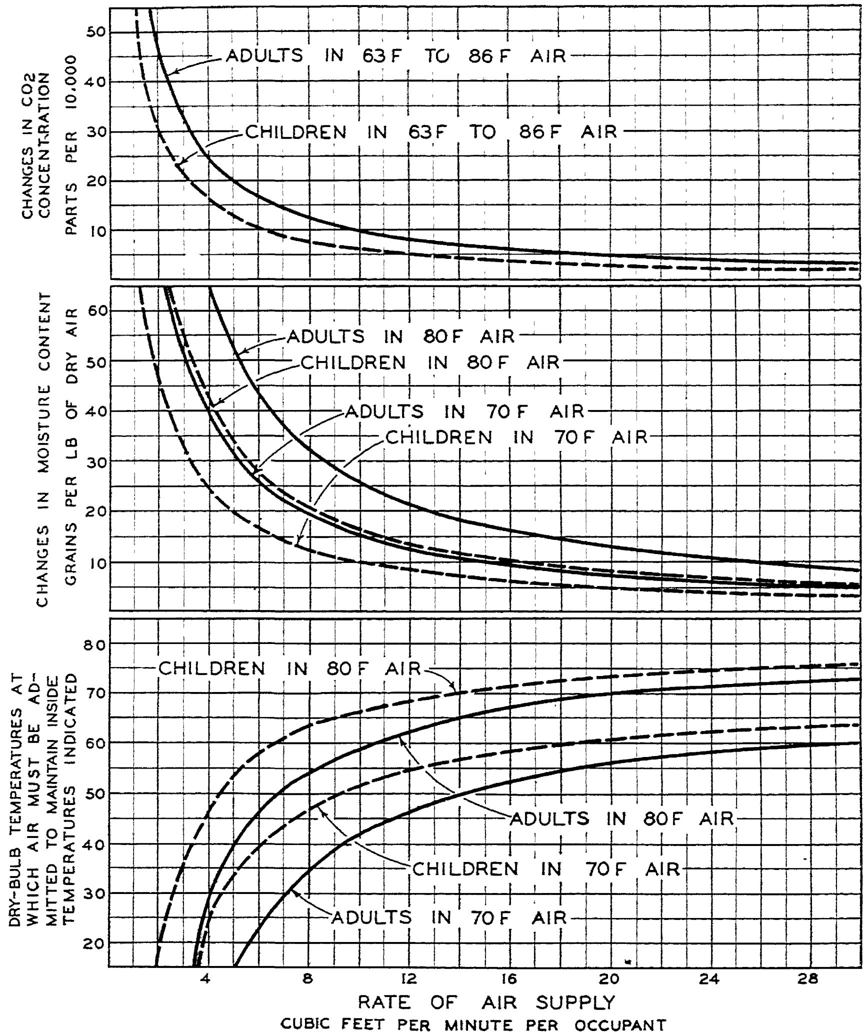

FIG. 4. RELATION AMONG RATE OF AIR CHANGE PER OCCUPANT, CARBON DIOXIDE CONCENTRATION AND MOISTURE CONTENT OF ENCLOSURE, AND DRY-BULB TEMPERATURE OF INCOMING AIR

Two of the most important factors on which the *temperature rise* depends are (1) the method of distribution and (2) the most economical temperature rise for the conditions involved. Some systems of distri-

bution produce drafts with but a few degrees temperature rise, while other systems operate successfully with a temperature rise as high as 35 deg. The total air quantity introduced in any particular case is inversely proportional to the temperature rise, and depends largely upon the judgment and ingenuity of the engineer in designing the most suitable system for the particular conditions. Small quantities of air reduce the size of equipment, ducts, space, and initial cost, but require lower air temperatures. In any specific case, the cost of refrigeration must be balanced against the extra cost in increased size of equipment and running expense.

Outside Air. In order to provide uniform temperature conditions, it is necessary to maintain a pressure of about 0.1 in. of water in the room or space to be ventilated or conditioned. This usually requires the introduction of a certain amount of outside air which depends on the particular conditions involved, and may vary over a considerable range.

In rooms in which the only source of contamination is the occupant the minimum quantity of outside or *new* air to be circulated appears to be that necessary to remove objectionable *body odors.* The concentration of body odors in turn depends largely upon the temperature of the air; the higher the temperature, the greater the amount of perspiration (sensible or insensible) given off from the skin, and the greater the concentration of odors.

NATURAL AND MECHANICAL VENTILATION

Under favorable conditions natural ventilation methods properly combined with means for heating may be sufficient to provide for the foregoing standards. As a rule, in instances in which the only source of contamination is the occupant, the requirements may be fulfilled when the following conditions prevail:

1. At least 50 sq ft of floor area for each occupant.
2. At least 500 cu ft of air space per occupant.
3. Effective openings in windows and skylights equal to at least 5 per cent of the floor area.

Whenever natural means are not sufficient to maintain the standards, resort must be made to whatever modifications or mechanical apparatus are necessary to secure such standards.

In large offices, large school rooms, and in public and industrial buildings, natural ventilation is uncertain and makes heating difficult. The chief disadvantage of natural methods is the lack of control: they depend largely on weather and upon the velocity and direction of the wind. Rooms on the windward side of a building may be difficult to heat and ventilate on account of drafts, while rooms on the leeward side may not receive an adequate amount of air from out of doors. The partial vacuum produced on the leeward side under the action of the wind may even reverse the flow of air so that the leeward half of the building has to take the *drift* of the air from the rooms of the windward half. Under such conditions no outdoor air would enter through a leeward window opening, but room air would pass out.

In warm weather natural methods of ventilation afford little or no

control of indoor temperature and humidity. Outdoor smoke, dust and noise constitute other limitations of natural methods.

RECIRCULATION

The saving in operating costs due to recirculation of the air, while very considerable, must not be obtained at the expense of air quality. The percentage of recirculated air may be varied to suit the seasonal changes so as to conserve heat in winter and refrigeration in summer, but at no time during occupancy should there be taken from out of doors less than 10 cfm for each occupant. As a general rule, recirculation impairs the quality of the air by excessive humidity (if not conditioned), excessive odors, or both, and it tends to deprive the air of its ionic content, but

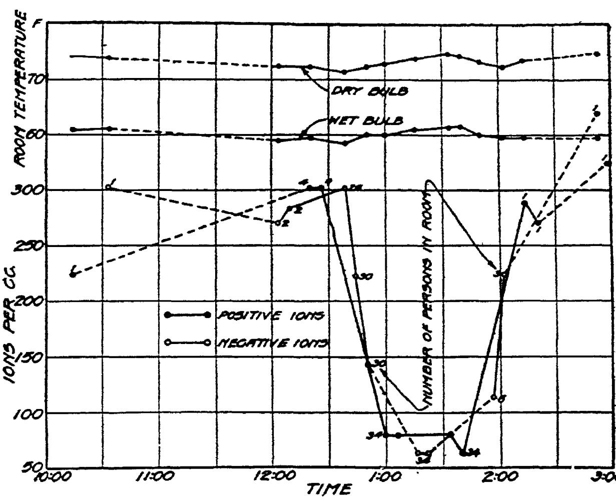

FIG. 5. INFLUENCE OF ROOM OCCUPANCY ON IONIC CONTENT[35]
(Cubical Contents of Room, 10,000 Cu Ft; Number of Occupants, 34)

the influence of this factor on comfort and health is at present a matter of speculation.

Toilets, kitchens, and similar rooms, in buildings using recirculation, should be separately mechanically ventilated by exhausting the air from them in order to prevent objectionable odors from diffusing into other parts of the building.

ULTRA-VIOLET RADIATION AND IONIZATION

In spite of the rapid advances made in the field of air conditioning during the past few years, the secret of reproducing, in indoor spaces, atmospheres of as stimulating qualities as those existing outdoors in the country, under ideal weather conditions, has not as yet been found. In fact, extensive studies have failed to elucidate the cause of the stimulating quality of outdoor country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical

TABLE 3. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[35]

Activity	Metabolic Rate Btu per Hour for Average Man (19.5 Sq Ft Surface Area)	Authority
Seated at rest	384	Research Laboratory, American Society of Heating and Ventilating Engineers.
Standing at rest	431	Research Laboratory, American Society of Heating and Ventilating Engineers.
Walking 2 mph	761	Average values from Douglas, Haldane, Henderson and Schneider; and Henderson and Haggard.
Walking 3 mph	1049	Douglas, Haldane, Henderson and Schneider
Walking 4 mph	1388	Average values from Douglas, Haldane, Henderson and Schneider; and Henderson and Haggard.
Walking 5 mph	2530	Douglas, Haldane, Henderson and Schneider
Slow run	2285	Henderson and Haggard
Very severe exercise	2555	Benedict and Carpenter
Maximum exertion	3333 to 4762+	Henderson and Haggard
Tailor	482	Becker and Hamalainen
Bookbinder	626	Becker and Hamalainen
Shoemaker	661	Becker and Hamalainen
Carpenter	762 to 963	Becker and Hamalainen
Metal worker	862	Becker and Hamalainen
Painter (of furniture)	876	Becker and Hamalainen
Stonemason	1488	Becker and Hamalainen
Man sawing wood	1797	Becker and Hamalainen

means. It is true that many suggestions have been advanced to account for the stimulating quality of outdoor air, such as ultra-violet light and ionization. At the present time neither of these suggestions has received any degree of scientific confirmation.

It is generally recognized that total outdoor solar radiation has marked curative value in certain diseases and is also a powerful germicidal agent. A critical review of the literature, however, does not substantiate the theory that ultra-violet radiation is of importance in air conditioning, since the use of ultra-violet sources fails to produce indoors, the previously mentioned stimulating qualities found in outdoor air.

Experiments[35] show that in occupied rooms there is a marked decrease in both positive and negative small ions. As shown in Fig. 5, soon after the occupants assembled the ionic content fell abruptly to a very low level which was maintained until the occupants left the room. Both positive and negative ions began to rise again as soon as the occupants departed.

The effects of the decrease in the ionic content of indoor air on comfort and health have not yet been subjected to sufficient scientific investigation. It would appear, however, from the evidence at hand, that comfort is not associated with a high ion content—but this must be considered, at least for the time being, as still a subject for further study.

[35]A.S.H.V.E. research paper entitled Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931). Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, August, 1933). Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin, (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1934). The Nature of Ions in Air and their Possible Physiological Effects, L. B. Loeb (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1934).

HEAT AND MOISTURE LOSSES

In order to solve air conditioning problems involving the human body it is necessary to know the rate at which sensible and latent heat are given up by the body under various conditions of temperatue and activity. Research at the A.S.H.V.E. Laboratory [32, 36] has resulted in the data given in Figs. 7, 8, and 9. Table 3 gives the metabolic rates for various degrees of activity.

The experimental data from which the curves were drawn show that

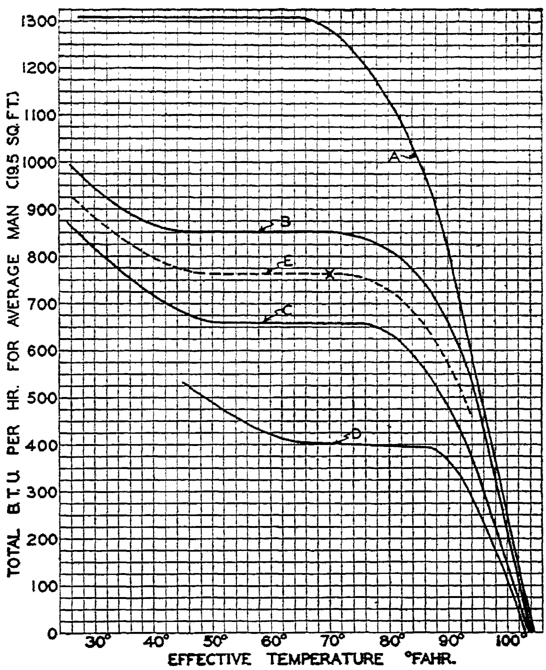

FIG. 6. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,160 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at an effective temperature of 70 deg only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

total heat loss does not vary appreciably within the comfort zone (see Fig. 6). Above or below this range the variation is approximately a function of effective temperature. Sensible and latent heat losses (Figs. 7 and 9) on the other hand, vary greatly within the comfort zone, the variation following closely the dry-bulb temperature.

Although total heat loss and sensible and latent heat losses are not exact functions of effective and dry-bulb temperature, respectively, for all

[36]Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (*American Journal of Hygiene*, Vol. XIII, No. 2, March, 1931, pp. 415-431).

conditions of humidity and air motion, they are plotted as such in the curves. This is accomplished by approximations which are sufficiently accurate for application to practical problems. Comparison of Figs. 7 and 8 shows how the cooling load may vary between sensible and latent heat elimination for different atmospheric conditions and activities of occupants.

An atmospheric condition resulting in sensible perspiration is to be

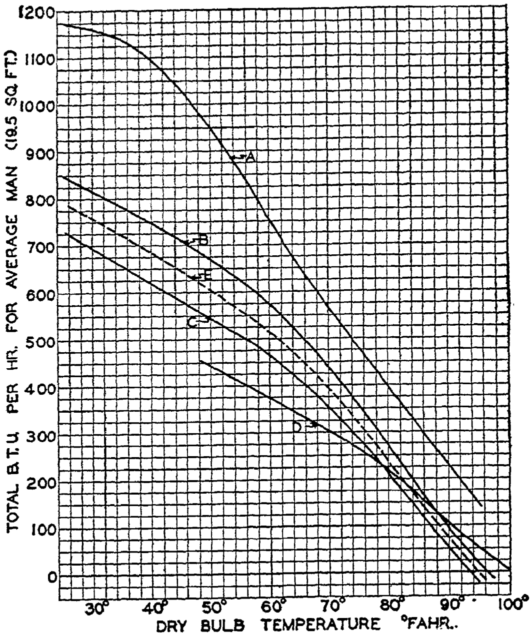

FIG. 7. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

avoided for obvious reasons. Tables 4 and 5 give the approximate effective temperatures at which perspiration is noticeable in different degrees for 95 per cent and 20 per cent relative humidity.

In theaters, auditoriums, department stores and other crowded enclosures, the amount of heat and moisture given off by the people is so large that normal changes in outside temperature and humidity have relatively little effect on indoor air conditions. The principal object of air conditioning in such places is to remove excessive heat and moisture by supplying a sufficient quantity of properly conditioned air. The indoor air conditions, however, must be varied according to the outside temperature, as has been pointed out.

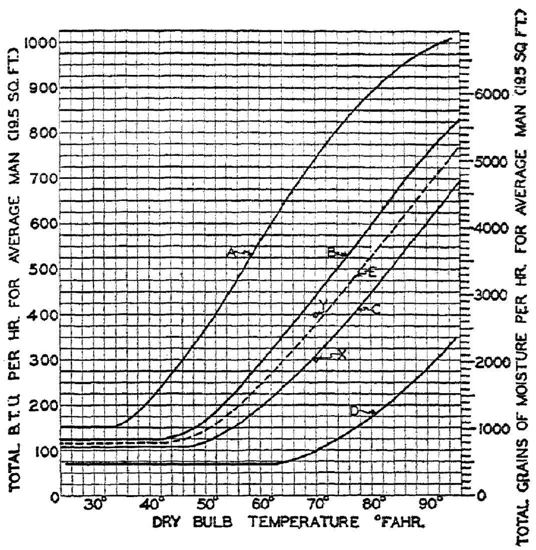

FIG. 8. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures.

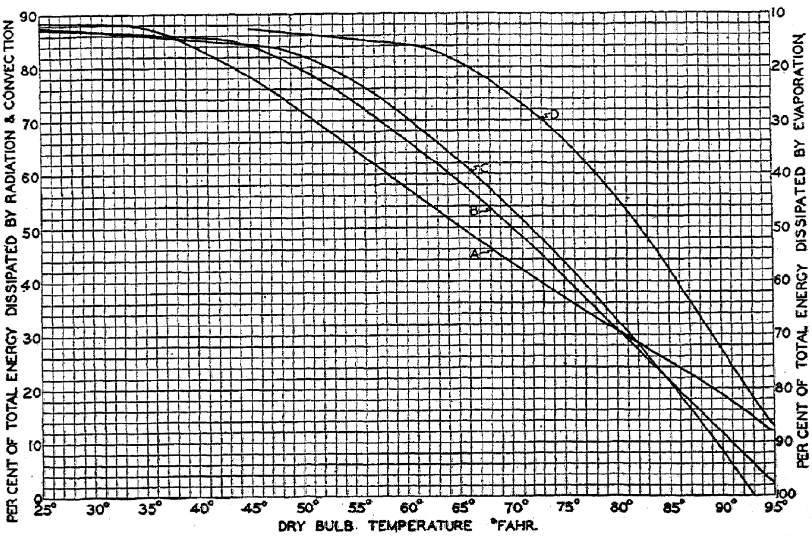

FIG. 9. HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves *B* and *D* which were drawn from data at many temperatures.

Although heat and moisture from the human body constitute the major portion of the cooling load, in most cases where air conditioning is provided for comfort and health other factors must also be considered. These include heat from lights, machinery, and processes, as well as the transmission and infiltration of heat through the building structure. The computations for these factors may be made in accordance with data given in Chapters 5 and 7.

TABLE 4. CONDITION OF SENSIBLE PERSPIRATION FOR PERSONS SEATED AT REST UNDER VARIOUS ATMOSPHERIC CONDITIONS[4]

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B	W. B.
Forehead clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body damp	**79.0**	79.7	78.4	**81.0**	97.5	67.5
Beads on forehead	**80.0**	80.8	79.4	**87.0**	109.4	75.2
Body wet	**84.5**	85.4	84.0	**86.5**	108.5	74.6
Perspiration on forehead runs and drips	**88.0**	89.0	87.6	**94.0**	125.2	85.4
Perspiration runs down body	**88.5**	89.5	88.1	**90.0**	116.0	79.5

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

TABLE 5. CONDITION OF SENSIBLE PERSPIRATION FOR PERSONS AT WORK UNDER VARIOUS ATMOSPHERIC CONDITIONS[30]

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B.
Forehead clammy	**59.0**	59.4	58.3	**69.5**	80.5	56.5
Body clammy	**50.0**	50.2	49.3	**57.0**	61.6	44.2
Body damp	**60.0**	60.3	59.3	**62.5**	69.6	49.5
Beads on forehead	**68.0**	68.5	67.5	**76.0**	91.0	63.4
Body wet	**69.0**	69.6	68.5	**71.0**	82.8	53.0
Perspiration on forehead runs and drips	**78.5**	79.3	78.0	**82.0**	100.5	70.2
Perspiration runs down body	**79.0**	79.8	78.5	**81.0**	99.8	69.0

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

In many cases, allowance must also be made for sun effect and for heat capacity of the building structure in accordance with studies by the A.S.H.V.E. Research Laboratory[37]. Another item to be considered is the radiant heat received by the body from high temperature wall and ceiling surfaces.

[37]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

PROBLEMS IN PRACTICE

1 ● What is the purpose and method of conditioning the air of occupied rooms?

Chiefly comfort, and the method is to control the temperature, humidity, and air distribution, and to prevent the accumulation of excessive body odors in the air. Other factors have yet to be studied.

2 ● What are the most comfortable air conditions?

Comfort standards are not absolute, but they are greatly affected by the physical condition of the individual, and the climate, season, age, sex, clothing, and physical activity. For the northeastern climate of the United States, the conditions which meet the requirements of the majority of people consist of temperatures between 68 and 72 F in winter and between 70 and 85 F in summer, the latter depending largely upon the prevailing outdoor temperature. The most desirable relative humidity range seems to be between 30 and 60 per cent.

3 ● Are the optimum conditions for comfort identical with those for health?

There are no absolute criteria of the prolonged effects of various air conditions on health. For the present it can be only inferred that bodily discomfort may be an indication of conditions that may produce poor health.

4 ● Given dry-bulb and wet-bulb temperatures of 76 F and 62 F, respectively, and an air velocity of 100 fpm, determine: (1) effective temperature of the condition; (2) effective temperature with still air; (3) cooling produced by the movement of the air; (4) velocity necessary to reduce the condition to 66 deg effective temperature.

(1) In Fig. 1 draw line *AB* through given dry- and wet-bulb temperatures. Its intersection with the 100-ft velocity curve gives 69 deg for the effective temperature of the condition. (2) Follow line *AB* to the right to its intersection with the 20-fpm velocity line, and read 70.4 deg for the effective temperature for this velocity or so-called still air. (3) The cooling produced by the movement of the air is $70.4 - 69 = 1.4$ deg effective temperature. (4) Follow line *AB* to the left until it crosses the 66 deg effective temperature line and interpolate velocity value of 340 fpm to which the movement of the air must be increased.

5 ● Given dry-bulb and wet-bulb temperatures of 75 and 68 F, respectively, first, what is the effective temperature? Second, is this condition warmer or cooler than 80 F dry-bulb and 60 F wet-bulb?

The first condition is given by the intersection of the 75 F dry-bulb line and the 68 F wet-bulb line (Fig. 3). The effective temperature of 72.1 deg is given by the numerical value of the effective temperature line passing through this point and indicated by the scale along the saturation curve. The second condition is given by the intersection of 80 F dry-bulb and 60 F wet-bulb and is 71.8 deg ET. It is therefore 0.3 deg ET cooler than the first condition.

6 ● Given 76 F dry-bulb and 61 F wet-bulb, how many degrees difference are there between this condition and the winter comfort line or 66 deg ET?

The effective temperature for this condition is given by the intersection of the 76-F dry-bulb and 61-F wet-bulb lines and is 70 deg ET, which is 4 deg ET warmer than the comfort line.

7 ● Assume that the design of an air conditioning system for a theater is to be based on an outdoor dry-bulb temperature of 95 F and a wet-bulb temperature of 78 F with an indoor relative humidity of 50 per cent. According to Table 2, the dry-bulb temperature in the auditorium should be 80 F. Estimate the sensible and latent heat given up per person.

The sensible heat given up per person per hour under this condition may be obtained from Fig. 7. With an abscissa value of 80 F, Curve *D* for men seated at rest gives a value on the ordinate scale of 220 Btu per person per hour as the sensible heat loss. The latent heat given up by a person seated at rest per hour may be obtained from Fig. 8. With an

abscissa value of 80 F, Curve *D* indicates a latent heat loss of 175 Btu per hour (left hand scale) or a moisture loss of 1190 grains per hour (right hand scale).

8 ● How much sensible heat, how much latent heat and how much water vapor will be added per hour to the atmosphere of an auditorium by an audience of 1000 adults, when the dry- and wet-bulb temperatures are 75 F and 63.5 F, respectively?

From Curve *D*, Fig, 7, find the sensible heat loss per person for a dry-bulb temperature of 75 F and still air to be 265 Btu per hour. From Fig. 8 find the latent heat loss per person for a dry-bulb temperature of 75 F to be 134 Btu per hour and the moisture added to be 905 grains per hour. Sensible heat = 1000 × 265 = 265,000 Btu. Latent heat = 1000 × 134 = 134,000 Btu. Water vapor added per hour to the air in the auditorium = 1000 × 905 = 905,000 grains or 129 lb.

The sensible and latent heat added to the air may also be found as follows: The effective temperature for dry- and wet-bulb temperatures of 75 F and 63.5 F, respectively, is 70.3 deg. From Curve *D*, Fig. 6, find 403 Btu as the total heat added to the air by a person for an effective temperature of 70.3 deg. From Fig. 9 find the percentage of sensible and latent heat at a dry-bulb temperature of 75 F to be 66.5 per cent and 33.5 per cent. The sensible heat added to the air in the auditorium is 1000 × 0.665 × 403 = 267,995 Btu per hour. The latent heat added is 1000 × 0.335 × 403 = 135,005 Btu per hour.

9 ● If the dry- and wet-bulb temperatures of the auditorium were 85 F and 63 F, respectively, how much heat and moisture would be dissipated to the atmosphere?

From Figs. 7 and 8, respectively, the sensible and latent heat losses per person for a dry-bulb temperature of 85 F are found to be 164 and 225 Btu per hour. The water vapor added to the atmosphere is 1520 grains per hour. The audience will then add 164,000 Btu sensible heat, 225,000 Btu latent heat and 1,520,000 grains or 217 lb of water vapor to the air in the auditorium per hour.

10 ● Neglecting the gain or loss of heat to an auditorium by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air, with dry- and wet-bulb temperatures of 65 F and 59 F, respectively, (63.1 deg ET) must be supplied per hour to an auditorium containing 1000 people in order that the inside shall not exceed 75 F (dry-bulb) and 65 F (wet-bulb), respectively?

Figs. 7 and 8 give 265 Btu sensible heat and 905 grains of moisture as the additions per person with a dry-bulb temperature of 75 F in the auditorium. Therefore, 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be required to raise the dry-bulb temperature from 65 to 75 F and $\frac{265,000}{2.4} = 110,400$ lb of air or 110,400 × 13.4 = 1,479,000 cfh of air will be required. This is equivalent to $\frac{1,479,000}{1000 \times 60} = 24.7$ cfm per person.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition is 65 grains. The increase in moisture content will therefore be 11 grains per pound of dry air. Hence $\frac{905,000}{11.0} =$ 82,300 lb of air at the specified condition will be required. This is equivalent to 82,300 × 13.4 = 1,103,000 cfh of air or $\frac{1,103,000}{1000 \times 60} = 18.4$ cfm of air per person.

The higher volume of 24.7 cfm per person will be required to keep the dry-bulb temperature from rising above the 75 F specified. The wet-bulb temperature will therefore not rise to the maximum of 65 F.

11 ● Assume that a man performs work at a rate equivalent to 50,000 ft-lb per hour, in an atmosphere having a dry-bulb temperature of 70 F. Estimate the sensible and latent heat given off per hour.

Since the net mechanical efficiency of the human body is about 20 per cent, the increase in metabolism due to work, over the resting metabolism, will be $\frac{50{,}000}{778^1 \times 0.20} = 320$ Btu per hour. Assuming a resting metabolism of 400 Btu per hour (see Fig. 6), the total metabolism during work will be $400 + 320 = 720$ Btu per hour, and the total heat loss $720 - \frac{50{,}000}{778} = 656$ Btu per hour, approximately. In Fig. 9, follow a vertical line from a dry-bulb temperature of 70 F to a point midway between Curves *A* and *B*. The sensible heat loss is about 46 per cent of the total loss, or $0.46 \times 656 = 302$ Btu per hour, and the latent heat is 54 per cent of the total or $0.54 \times 656 = 354$ Btu per hour.

12 ● The characteristics of air supplied to ventilate a room are:

Carbon dioxide concentration	4 parts per 10,000
Wet-bulb temperature	45.2 F
Dry-bulb temperature	55.0 F
Moisture content	29.0 grains per pound of dry air

a. What will be the dry-bulb temperature of the air in the room if it is occupied by five adults, if the air change, including both ventilation and infiltration, is 50 cu ft per minute, and assuming that there is no heat gain or loss to the room from any source other than from the occupants?

b. What will be the carbon dioxide concentration of the air in the room under these conditions?

c. What will be the moisture content of the air in the room under these conditions?

d. What will be the wet-bulb temperature and the relative humidity of the air in the room under these conditions?

e. What would the temperature of the incoming air have to be to give a room a dry-bulb temperature of 70 F?

a. The air change is 10 cu ft per minute per occupant. From the bottom chart of Fig. 4 at the intersection of an incoming air dry-bulb temperature of 55.0 F and a rate of air supply of 10 cu ft per minute per occupant, find by interpolation between the 70 F and 80 F adult curves the dry-bulb temperature of the air in the room to be 78.0 F.

b. From the top chart of Fig. 4 find the increase in CO_2 concentration to be 10 parts of CO_2 per 10,000 parts of air. Therefore, the air in the occupied room will contain 14 parts of CO_2 per 10,000.

c. From the center chart in Fig. 4 find by interpolation between the 70 F and 80 F adult curves the increase in moisture content to be 23 grains per pound of dry air for adults in 78 F air. This gives a resultant moisture content of the air in the room of 52 grains per pound of dry air.

d. From the psychrometric chart, Fig. 3, find the resulting wet-bulb temperature and relative humidity for 78 F dry-bulb and 52 grains of moisture to be 61.0 F and 37 per cent, respectively.

e. From the bottom chart, Fig. 4, find the required incoming air temperature to be 42 F dry-bulb.

13 ● Name three factors that influence the feeling of warmth and the elimination of body heat.

Temperature, humidity, and air movement.

14 ● What is meant by effective temperature?

Effective temperature is a composite index which combines the measurements of temperature, air motion, and humidity into a single value. It is an arbitrary index of the degree of warmth or cold felt by the human body due to these factors.

15 ● Referring to the A.S.H.V.E. Comfort Chart (Fig. 3), list the conditions (dry-bulb, wet-bulb, effective temperature, and humidity) which will produce comfort at each corner of the average winter comfort zone and of the average summer comfort zone.

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[1]Heat equivalent of mechanical work in foot-pounds per Btu.

Average winter comfort zone:

Wet-Bulb F	Dry-Bulb F	Relative Humidity
58.5	64.5	70 per cent
51.5	67.5	30 per cent
59.0	79.0	30 per cent
67.0	74.0	70 per cent

Average summer comfort zone:

Wet-Bulb F	Dry-Bulb F	Relative Humidity
62.0	68.0	70 per cent
54.0	72.0	30 per cent
63.5	85.0	30 per cent
71.5	78.5	70 per cent

16 ● What is generally considered to be the desirable and practicable range of relative humidity indoors?

30 per cent to 60 per cent.

Chapter 3

INDUSTRIAL AIR CONDITIONING

Moisture Content and Regain, Hygroscopic Materials, Atmospheric Conditions Required, Air Conditioning of Libraries, Banana Ripening, Lumber Drying, Greenhouse Heating, Apparatus for Industrial Conditioning

AIR conditioning is applicable to industrial or process conditioning for the improvement of products during manufacture, or for making the process independent of climatic conditions. In many industries, the temperature and relative humidity of the air have a marked influence upon the rate of production and the weight, strength, appearance, and general quality of the product. These results are due to the fact that most materials of animal or vegetable origin, and to a lesser extent minerals in certain forms, either take up or give moisture to the surrounding air.

MOISTURE CONTENT AND REGAIN

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not necessarily imply that the material as a whole has been completely dried out and has re-absorbed moisture. In the case of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. In measuring moisture it is necessary to dry out a sample so that the loss in weight may be used as a basis for calculating the regain of the whole lot.

TABLE 1. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

CLASSIFICATION	MATERIAL	DESCRIPTION	RELATIVE HUMIDITY—PER CENT 10	20	30	40	50	60	70	80	90	AUTHORITY
Natural Textile Fibres	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Misc. Organic Materials	Leather	Sole oak—tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fibre	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

HYGROSCOPIC MATERIALS

Air conditioning is extensively used in the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco, and foodstuffs. Where the physical properties of the product affect value, the question of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Economy of manufacturing, therefore, requires that the moisture content be maintained at a percentage most favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A constant condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material, and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight it is proper that they contain a normal or standard moisture content. Air conditioning is important in certain branches of the chemical industry in controlling the temperature of reaction and facilitating or retarding evaporation. The control of moisture content of air supplied to blast furnaces in the manufacture of pig iron also has proved advantageous.

The moisture content of a hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

Table 1 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

Conditioning and Drying

Exposure of hygroscopic materials to an atmosphere of controlled humidity and temperature for the purpose of establishing a specified moisture condition in the material is called *conditioning*. Where the desired final moisture content is relatively low, the term *drying* is usually used. In any case, control of relative humidity, temperature, air velocity and length of exposure are all of more or less importance.

The conditioning treatment may be undertaken in a special enclosure (conditioning room) or it may be accomplished in the same room and at the same time as some regular manufacturing process. For instance, in the weaving of textiles a high relative humidity is commonly employed to keep the yarn strong and pliable, thus assisting in the weaving process and

Table 2. Desirable Temperatures and Humidities for Industrial Processing

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Automobile	Assembly line	65	40
Baking	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
Biological Products	Vaccines	below 32	
	Antitoxins	38 to 42	
Brewing	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
Ceramic	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
Chemical	General storage	60 to 80	35 to 50
Confectionery	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
Distillery	General manufacture	60	45
	Storage of grains	60	30 to 45
Drug	Storage of powders and tablets	70 to 80	30 to 35
Electrical	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
Food	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35
Fur	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 2. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING (Continued)

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
INCUBATORS	Chicken	99 to 102	55 to 75
LABORATORY	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
LEATHER	Drying of hides	90	
LIBRARY	Book storage (see discussion in this chapter)	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCH	Manufacturing	72 to 74	50
	Storage of matches	60	
MUNITIONS	Fuse loading	70	55
PAINT	Drying of lacquers	60 to 80	25 to 50
	Drying of oil paints	60 to 90	25 to 50
	Brush and spray painting	60 to 80	25 to 50
PAPER	Binding, cutting, drying, folding, gluing	60 to 80	25 to 50
	Storage of paper	60 to 80	35 to 45
PHOTOGRAPHIC	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
PRINTING	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	60 to 75	20 to 60
	Storage of rollers	60 to 80	35 to 45
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	60 to 70
	weaving	68 to 75	70 to 80
	Rayon— spinning	70	85
	twisting	70	65
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
TOBACCO	Cigar and cigarette making	70 to 75	55 to 65
	Softening	90	85
	Stemming or stripping	75 to 85	70

at the same time leaving the product in a satisfactory condition of regain for commercial reasons.

As a rule, commercial regain standards are specified percentages which by test have been found equivalent to a so-called *standard atmosphere* with which the goods would be in hygroscopic equilibrium after prolonged exposure. Committee D13 on Textiles of the *American Society for Testing Materials* has adopted a relative humidity of 64 to 66 per cent and a temperature of 70 to 80 F as the standard atmosphere for textile testing.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process represents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions and controlling them automatically. Also, in departments where people are working, their health, comfort, and productive efficiency must be considered. A compromise often is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 2 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

AIR CONDITIONING OF LIBRARIES[1]

Temperature has little effect on the preservation of books. A temperature over 100 F, combined with low relative humidity, may cause the book materials to become brittle, while a temperature much below freezing may cause permanent deterioration of the glue in the binding. The relative humidity should be maintained between 40 and 70 per cent, although these limits need not hold for short periods of time. If the relative humidity gets much below 40 per cent, first the glue and then the paper will tend to become brittle which will not cause any permanent damage unless the book is used while in this condition, as a subsequent increase in humidity will bring the materials back to their normal condition. If the relative humidity gets above 80 per cent, the growth of mildew may be expected.

One of the principal agents of destruction and deterioration of paper and books in libraries is sulphur dioxide gas in the air. If air containing

[1]See *U. S. Bureau of Standards Bulletin No.* 128 entitled A Survey of Storage Conditions in Libraries, by Kimberly and Hicks.

sulphur dioxide is allowed to come in contact with cellulose, the principal constituent of paper, sulphuric acid is formed on the surface. This acid is not volatile at ordinary temperatures and therefore accumulates throughout the life of the paper. The destructive effect of the acid on the paper is independent of the relative humidity of the surrounding air. Low alkaline concentration spray water may be used in an air washer to neutralize the acid condition. Such an air washer must be especially constructed to resist corrosion.

BANANA RIPENING

Ripe bananas are very perishable and for this reason men who deal in them must depend mainly upon control of the ripening speed as a means of regulating their daily supply of the fruit. Knowledge and experience are required in regulating the ripening treatment and to control the ripening speed. An accurate appraisal must be based upon a careful examination of the fruit when received to determine its condition, and periodically, thereafter, to determine the rate of ripening.

Fast ripening may be accomplished in from three to four days after the green fruit is placed in a ripening room by adjusting the temperatures of the room until the pulp temperature reaches about 70 F. In warming up cool fruit, quick heating is recommended, and it is good practice to use sufficient heat to raise the average fruit temperature at the rate of 2 to 3 deg per hour. After the first 24 hours, the room should be held at 68 F until the fruit is colored and then reduced to 66 F and held at this temperature. A high relative humidity of from 90 to 95 per cent should be maintained until the bananas show color, when it may be reduced to about 80 per cent. High humidity is important during the warming period. No ventilation should be used until the fruit has colored, after which ventilation at a rate not to exceed four changes per hour may be used to assist in reducing the humidity and to freshen the air in the room. If the fruit shows slow or uneven ripening characteristics, one or two applications of ethylene gas of approximately 1 cu ft per 1000 cu ft of room space may be used.

Medium speed ripening of bananas in from five to seven days may be accomplished by holding the fruit at 64 F. The humidity and ventilation control should be the same as for fast ripening. A treatment with ethylene gas will seldom be necessary. For slow ripening in from nine to ten days, the fruit should be held at from 60 to 62 F. Temperatures below 62 F are not advisable for very thin fruit. The humidity should be the same as for fast ripening, and ventilation (up to 3 or 4 air changes per hour) should be used provided the humidity can be maintained. Ethylene gas treatment will not be required.

For holding ripened bananas, temperatures between 56 and 60 F are recommended. A reduction in humidity is beneficial in toughening the peel and reducing the mould, but too low a humidity will cause shrinkage. Although exact humidity control is not essential, the desirable range is between 75 and 80 per cent.

LUMBER DRYING

The United States Forest Products Laboratory, Madison, Wis., has

prepared eleven schedules[2] for the kiln-drying of practically all kinds, types, and thicknesses of softwoods and hardwoods. The tables given in these schedules range from 105 to 200 F dry-bulb, and from 20 to 80 per cent relative humidity. As a rule, the softer the wood, the higher the average temperature used. The temperature and relative humidity in a lumber drying kiln are varied for all conditions, starting with a low dry-bulb and a high relative humidity when the green lumber, containing a large percentage of moisture, is started to dry. As the moisture content of the lumber decreases, the dry-bulb temperature of the kiln is increased, and the relative humidity reduced. It is noted, however, that perfect drying does not necessarily result from following a schedule, and that an operator must be trained to watch the condition of the stock in the kiln and to immediately apply a remedy if he sees things going wrong.

GREENHOUSES

Table 3 lists customary dry-bulb temperature ranges for different types of plants and flowers raised in greenhouses.

TABLE 3. CUSTOMARY TEMPERATURES FOR DIFFERENT TYPES OF GREENHOUSES

TYPE OF HOUSE	TEMPERATURE RANGE DEG FAHR	TYPE OF HOUSE	TEMPERATURE RANGE DEG FAHR
Carnation	45 to 55	Orchid, cool	50 to 55
Conservatory (general collection)	60 to 65	Palm, warm	60 to 65
Cool	45 to 50	Palm, cool	50 to 55
Cucumber	65 to 70	Propagating	55 to 60
Fern	60 to 65	Rose	55 to 60
Forcing	60 to 65	Sweet pea	45 to 50
General purpose	55 to 60	Tomato	65 to 70
Lettuce	40 to 45	Tropical	65 to 70
Orchid, warm	65 to 70	Violet	40 to 45

APPARATUS FOR INDUSTRIAL CONDITIONING

Apparatus for industrial air conditioning may be divided into two distinct groups, namely, (1) *humidifiers* for increasing the moisture content of the air and for producing cooling by evaporation and (2) *dehumidifiers* for removing moisture from the air and for producing cooling by contact with water or surfaces at a lower temperature than the air.

Strictly speaking, humidity control alone, whether it involves humidification or dehumidification, is not *air conditioning*. To be entitled to this classification according to the definition in Chapter 41, the process should include the *simultaneous control* of temperature, humidity and air motion.

Industrial humidifiers may be divided into the following general types, according to the method of operation:

1. *Direct*, which spray into the room.
2. *Indirect*, which introduce moistened air.
3. *Combined* direct and indirect.

[2]*Technical Note Number 175*, Forest Products Laboratory, U. S. Forest Service, Madison, Wis.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet.* Where distribution is obtained by *induction,* the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray which cannot be permitted when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled to the wet-bulb temperature. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is quickly evaporated and the resulting vapor is rapidly and thoroughly diffused. This effective distribution of fine spray over the maximum possible area insures complete and extremely rapid vaporization even at the highest humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

The spray and high-duty types of humidifiers have many features in common but the latter, because of its finer spray and greater capacity, is often considered better adapted for producing high humidities.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

Humidifiers and air washers are also described in Chapter 11.

Where large quantities of power are generated in a limited space and where a comparatively high relative humidity is required, it is often feasible and economical to use a combination of direct and indirect humidification. The indirect humidification provides the desired quantity of ventilation and cooling, and the additional direct humidification provides for increase in humidity without interfering with the ventilation or the cooling effected by the indirect system.

In general, it may be stated that direct humidification is most satisfactory where high humidities are desired but where little cooling, ventilation or air motion is required. Therefore, the indirect system is most applicable where either low or high relative humidities are desired with maximum cooling and ventilation effect. For conditions that require an unusually large amount of heat to be absorbed by ventilation, together with the maintenance of high humidities, it is often preferable to make use of the combination system of indirect and direct humidification. If the indirect system alone were used it would mean an unusually large volume of air to be handled, which might interfere, due to air motion, with production, even though it would result in greater cooling effect. If direct humidification alone were used, no ventilation would be obtained, with consequently higher room temperatures.

Dehumidifiers, which are similar in design and appearance to indirect humidifiers and air washers, are described in Chapter 11. The main differences are found in the internal construction of the dehumidifier, in

the use of refrigeration or of heat as required for controlling the water temperature, and in differences in the general methods of control.

PROBLEMS IN PRACTICE

1 ● A condition of 75 F dry-bulb temperature and 55 per cent relative humidity is being maintained in a cigarette manufacturing department. What will be the regain and moisture content of the tobacco?

The regain, from Table 1 = 17.75 per cent.

$$\text{The moisture content} = \frac{17.75 \times 100}{100 + 17.75} = 15.1 \text{ per cent.}$$

2 ● A 1-lb sample taken from a 100-lb batch of material is found to have a bone dry weight of 0.89 lb. This material is to be processed under atmospheric conditions which should produce a regain of 15 per cent. Compute the finished weight for each original 100-lb batch.

Let W equal the number of pounds of moisture in a finished batch.

$$\frac{W}{89} = \text{regain} = 15 \text{ per cent} = \frac{15}{100}$$

$$W = 13.35$$

$$89 + 13.35 = 102.35 \text{ lb finished weight.}$$

3 ● A bundle of sea island cotton is found to have a bone dry weight of 9.26 lb. What is the proper relative humidity at 75 F to produce a weight of 10 lb at equilibrium?

Desired conditioned weight = 10.00 lb
Bone dry weight = 9.26 lb

Weight of moisture required = 0.74 lb

$$\text{Regain} = \frac{0.74}{9.26} \times 100 = 7.9 \text{ per cent.}$$

From Table 1, the proper relative humidity required is 60 per cent.

4 ● Compute the bone dry weight of 1000 lb of manila rope which has been stored for a considerable period of time in a conditioned room at 75 F dry-bulb temperature and 50 per cent relative humidity.

Assuming that this material has come to equilibrium under the atmospheric conditions given, Table 1 shows a regain of 8.5 per cent.

Let W equal the total weight of moisture in pounds.

$1000 - W$ = bone dry weight in pounds.

$$\frac{W}{1000 - W} = \text{regain} = 8.5 \text{ per cent} = \frac{8.5}{100}$$

$$W = 78.3 \text{ lb moisture}$$

$$1000 - 78.3 = 921.7 \text{ lb bone dry weight.}$$

5 ● An egg evaporating plant wishes to dry 2000 lb of egg whites (85 per cent water) to crystalline form each 24 hours. The maximum permissible air delivery temperature in the dryer is 140 F. What air volume will be required, assuming that outside air is at 95 F dry-bulb and 78 F wet-bulb and that air leaves the dryer 70 per cent saturated?

Moisture to be removed = 2000 × 0.85 = 1700 lb. Using psychrometric chart and starting at the intersection of the vertical 95 F dry-bulb temperature line and the 45 per cent humidity line, move horizontally to the right to the intersection with the 140 F vertical temperature line at 10 per cent relative humidity; then move along the constant

heat (or wet-bulb line) to its intersection with the 70 per cent relative humidity curve and read 94 F dry-bulb, which will be the temperature of the air leaving the dryer.

Moisture per cubic foot at 94 F and 70 per cent relative humidity	=	11.8 grains
Moisture per cubic foot at 95 F and 78 F wet-bulb	=	8.0 grains
Moisture added per cubic foot of air handled	=	3.8 grains

$$\frac{1700 \times 7000}{24 \times 60 \times 3.8} = 2170 \text{ cfm.}$$

No allowance is made for heat lost in the transmission to and from the dryer or for the heat required to raise the product from its entering temperature to that maintained in the dryer. This would necessitate a trial and error solution common to all drying problems.

6 ● It is proposed to install a central fan type air conditioning system comprised of fan, air washer, filters, and heating coils to provide ventilation and to maintain proper humidity in a small library during periods of winter operation. The heat loss has been estimated at 450,000 Btu per hour in maintaining a condition of 72 F dry-bulb and 45 per cent relative humidity. Assuming that the air washer completely saturates the air, what must be the leaving dry-and wet-bulb temperatures to provide the required condition?

49.85 F is the dew-point temperature corresponding to the stated required condition.

7 ● Assuming a maximum permissible air delivery temperature of 100 F in Question 6, what air volume will be required?

$$\frac{450{,}000 \times 55.2}{(100 - 72) \times 60} = 14{,}800 \text{ cfm.}$$

8 ● If in Questions 6 and 7 it is assumed that winter humidity control will consist simply of a dew-point thermostat at the exit of the air washer, controlling the dew-point temperature by operating automatic dampers, and thereby proportioning the respective volumes of outside and recirculated air admitted:

a. What volume of air should be recirculated?

b. What volume of air will be exfiltrated from the buildings?

c. What reheating capacity will be required?

a. Btu per pound at 72 F and 45 per cent relative humidity	=	25.38
Btu per pound at 0 F (assumed saturated)	=	0.85
Btu per pound at 49.85 F saturated	=	20.11

$$\text{Recirculated air} = \frac{(20.11 - 0.85)}{(25.38 - 0.85)} \times 14{,}800 = 11{,}600 \text{ cfm.}$$

b. The same volume as is introduced as fresh outside air, namely,

$$14{,}800 - 11{,}600 = 3200 \text{ cfm.}$$

c. The reheaters must be of such capacity as to reheat the volume of air handled from 49.85 (the dew-point) to 100 F.

$$\frac{14{,}800 \times (100 - 49.85) \times 60}{55.2} = 808{,}000 \text{ Btu per hour.}$$

Chapter 4

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases with mechanical forces, finds extensive application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are the wind and the difference in temperature of the air inside and outside the building. The arrangement and control of ventilating openings should be such that the two forces act coöperatively and not in opposition.

Wind Forces

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings and trees.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \qquad (1)$$

where

Q = air flow in cubic feet per minute.
A = free area of inlet (or outlet) openings in square feet.
V = wind velocity in feet per minute,
= miles per hour × 88.
E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle.)

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E.

The accuracy of the results obtained by the use of Formula 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edge orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less, and if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following four places:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. In a monitor on the side opposite from the wind.
4. In roof ventilators or stacks exposed to the full force of the wind[1].

Forces due to Stack Effect[2]

The stack effect produced within a building is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\,\sqrt{H\,(t_1 - t_2)} \tag{2}$$

where

Q = air flow in cubic feet per minute.
A = free area of inlets or outlets (assumed equal) in square feet.
H = height from inlets to outlets, in feet.
t_1 = *average* temperature of indoor air in height H, in degrees Fahrenheit.
t_2 = temperature of outdoor air, in degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

The height between inlets and outlets should be the maximum which the building construction will allow.

In some cases the necessary air flow will be known from the requirements of the building occupancy, and the area necessary for certain assumed temperature differences may be calculated. Or the areas may be fixed by the building construction, and the maximum air flow for various differences between indoor and outdoor temperatures may be calculated. In any case, the conditions which give the minimum air flow are those which control the design, as the system must have ample capacity even under the most unfavorable conditions which are those of mild or warm weather.

TYPES OF OPENINGS

The engineering problems of a natural ventilation system consist of the *design, location, and control of ventilating openings* to best utilize the

[1]See Airation of Industrial Buildings, by W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

[2]See Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926), and Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

natural ventilation forces, in accordance with the requirements of building occupancy. The types of openings may be classified as:

1. Windows, doors, monitor openings, and skylights.
2. Roof ventilators.
3. Stacks connecting to registers.
4. Specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in

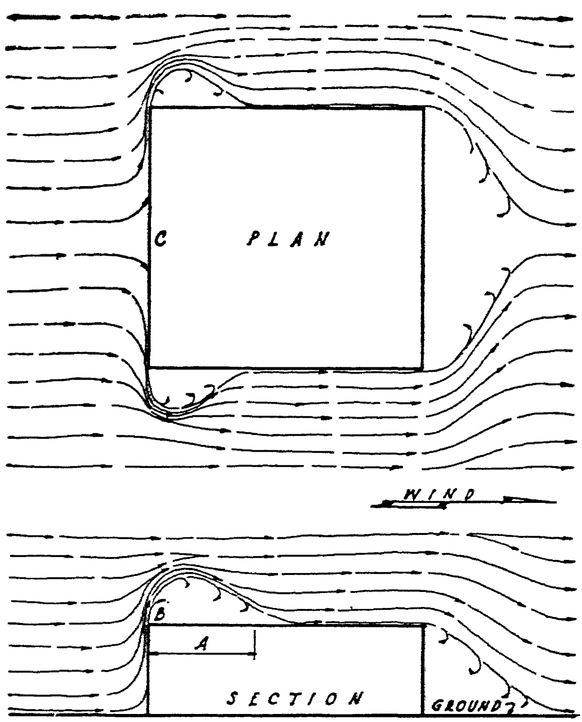


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (A—LENGTH OF SUCTION AREA; B—POINT OF MAXIMUM INTENSITY OF SUCTION; C—POINT OF MAXIMUM PRESSURE)

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top or bottom. Whatever the form and type of window used, the amount of clear area that can be made available is the factor of greatest importance in ventilation.

All types of sash (double-hung, top, center or bottom horizontal pivoted, or vertical pivoted) have about the same air flow capacity for the same clear area. Air leakage through *closed* windows is important during high winds (Chapter 6).

The proper distribution of air in occupied spaces is an element almost as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Door openings are seldom included in the ventilation calculations, though they may be of great value for extreme summer conditions, and should be considered in this connection as well as in garage design.

Skylight and monitor openings are of importance as these and the roof ventilators are outlets, while the lower windows are usually inlets on the windward side and outlets on the leeward side. In general the areas of inlets and of outlets should be about equal. It is important to make a check on this ratio in any installation, as any great excess of area of one set of openings over another means waste opening area. The operating devices used for sash, monitors, skylights and roof ventilators should be well selected as poor operating devices may defeat the entire design.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet, which is sensitive to wind action for producing additional flow capacity, and at the same time is subject to manual or automatic control by suitable dampers. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it offers to air flow, (3) the area and location of openings provided for air inflow at a lower level, and (4) the ability of the ventilator head to utilize the kinetic energy of the wind for inducing flow by centrifugal or ejector action. Frequently one or more of these capacity factors is overlooked in a ventilator installation.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1.) This does not mean that no ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little importance in these locations, and hence their size should be increased proportionally.

Ventilator resistance depends on (1) type of inlet, (2) area of openings and passages, and (3) number of turns or changes of direction of the air flow. The inlet grille, if any, should have ample free area, and the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of a square-entrance orifice (flow coefficient 0.60). In other words, the grilles should be oversize as compared with the ventilator, and they should be connected by tapering collars. If the ventilator head construction produces changes in the direction of air flow, the area of the flow passages should be increased accordingly.

Air inlet openings at lower levels in the building are of course necessary for the economical use of ventilator capacity. The inlet openings should be at least equal to, and preferably twice as great as the combined throat areas of all roof ventilators. The air discharged by a roof ventilator

depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 11. These may be classified as *stationary*, Figs. 2 to 6, *pivoted* or *oscillating*, Figs. 7 to 9, or *rotating*, Figs. 10 and 11. When selecting unit ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

It should be kept in mind that a suitable combination of roof ventilators with mechanical ventilation frequently offers the best solution of a ventilating problem. The natural ventilation units may be used to supplement power driven supply fans, and under favorable weather conditions it may be possible to shut down the power driven units. Where low operating costs are very important, such a combination has great advantages. Roof ventilators with built-in electric fans are attracting increased attention because they combine the advantages of low installation and operating cost with those of continuous service.

Controls

In connection with any combination between natural and fan ventilation, the controls are of importance. Both the fans and the ventilator dampers may be controlled by some combination of three methods: (1) hand operation, (2) thermostat operation, and (3) control by wind velocity. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called *gravity* action). While their openings projecting above the roof are not provided with any special construction for developing suction by the action of the wind, the plain vertical opening is also effective in this respect. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

Stacks are applicable particularly in the case of schools, apartments, residences and small office buildings. Partitions interfere with general air circulation, and some type of outlet from each room is necessary. If the building is not too tall, and the requirements of occupancy are moderate, a system of stacks with registers in each room may be more economical than a system of mechanical ventilation employing fans. In making the comparison, however, the building space occupied by the stacks should be considered.

With little or no wind, chimney effect or temperature difference will produce outflow through the stacks and an equal inflow through windows

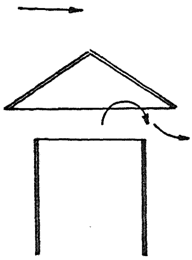


FIG. 2

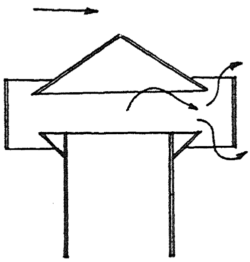


FIG. 3

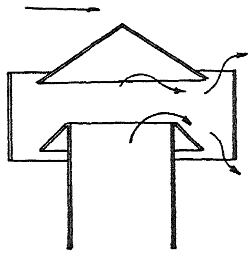


FIG. 4

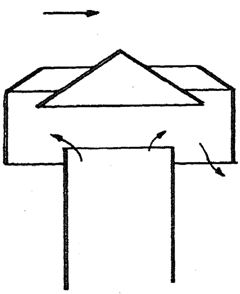


FIG. 5

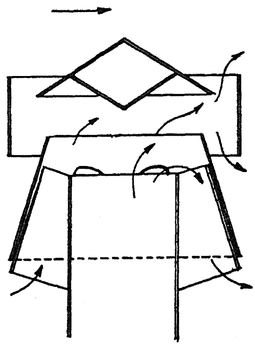


FIG. 6

SIX COMMON TYPES OF STATIONARY VENTILATORS

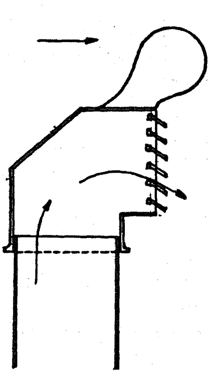


FIG. 7

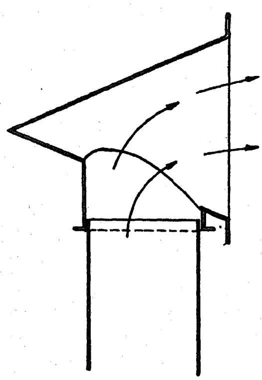


FIG. 8

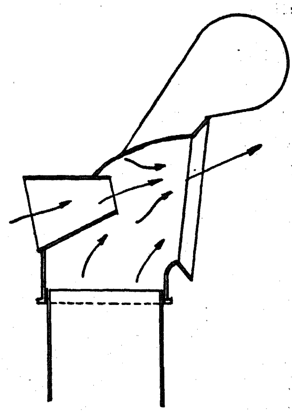


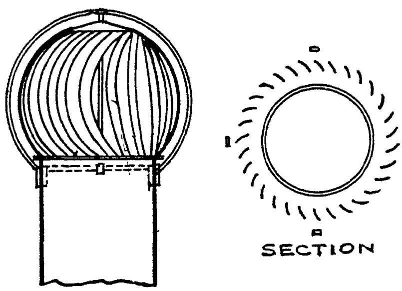
FIG. 9

THREE TYPICAL OSCILLATING VENTILATORS

in all sides of the building. With wind, the inductive force at the top of ventilating shafts is more powerful than that on the leeward side of the building, so that air is drawn in through leeward openings by a combination of the forces of wind and temperature difference. On the windward side, the direct forcing pressure of the wind is of course added to the temperature difference effect. Thus forces are available for causing inflow at practically every window of such a building. Adequacy of stack size must, of course, be provided.

PRINCIPLES OF AIR FLOW CONTROL

The air flow through a ventilation opening depends on the two factors already discussed, namely, (1) the natural forces available, (2) the openings available, and the resistance to flow offered by these openings. The design problem includes, of course, a determination of the *desired air*



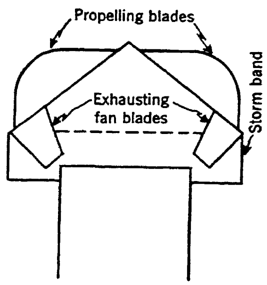


FIG. 10. ROTATING VENTILATORS FIG. 11.

quantity and distribution in order that the openings may be properly placed.

The purpose of ventilation is to carry off either excess heat or air impurities, and the desired air quantities depend upon the amount of heat or of impurities present. The amount of heat can be determined, in the case of forge shops for example, from the amount of fuel burned, which in turn is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some instances, not all of the heat may be dissipated to the air, but a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$H = \frac{c\,Q\,D}{V} \tag{3}$$

where

c = 0.24 = specific heat of air.

V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 41.)

H = heat to be carried off, in Btu per minute.
Q = air flow in cubic feet per minute.
D = inlet-outlet temperature difference in degrees Fahrenheit.

For disposing of air impurities, the required air flow must be such that the outside air will dilute the impurities to a degree that they are no longer objectionable. For human occupancy, such as in auditoriums and classrooms, 10 cfm per person is usually taken as the minimum of outside air necessary for ventilation (see Chapter 2). For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Air *quantity* and *quality* are not the only requirements. For human occupancy, air *distribution* is important. In ventilation the air distribution is almost entirely a matter of the number, the design, and the location of inlets and outlets. In locating openings, special precautions should be taken against the formation of dead air spaces or *pockets* within the zone of occupancy.

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two estimated quantities. The same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity[3], this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations[1, 2] show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicted with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Temperature differences are 10 F in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per minute is:

$$H = \frac{15}{60} \times 7.75 \times 18{,}000 = 34{,}875 \text{ Btu.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{cD} = \frac{13.5 \times 34{,}875}{0.24 \times 10} = 196{,}172 \text{ cfm.}$$

[3]This is true for *turbulent* flow only. It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings.

This is equal to 19.6 air changes per hour. The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening. Actually, the latter difference is larger and so the value of 19.6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2:

$$A = \frac{Q}{9.4\sqrt{H\,(t_1 - t_2)}} = \frac{196{,}172}{9.4\sqrt{30 \times 10}} = 1{,}205 \text{ sq ft.}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0.50 \times 1{,}205 \times 5 \times 88 = 265{,}100 \text{ cfm.}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified. Apparently 1205 sq ft are a very generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example it must be apparent that while formulas may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulas.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone of occupancy.

2. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.

4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

5. In an industrial building where furnaces, that give off heat and fumes, are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then coöperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

6. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to coöperate with the wind.

7. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in coöperation of wind and temperature difference.

8. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

9. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

10. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. On the other hand, where the direction of the wind is quite variable, it may be stated as a general principle that windows should be arranged in sidewalls and monitors so that there will be approximately equal area on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force of the wind, and others opposed to a suction force, and effective movement through the building will be assured.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. Special consideration should be given to the possibility of sidewall or monitor windows being closed on account of weather conditions. Such possibilities favor roof ventilators and specially designed stormproof inlets.

MEASUREMENT OF NATURAL AIR FLOW

The determination of the performance of any ventilating system involves measurements which are not easy to make. The difficulties are increased in the case of natural ventilation, since the motive forces and the air velocities are very small. The measurements necessary for giving the *capacity* of a system are (1) velocity of the wind, (2) velocity of the air through inlet and outlet openings, (3) outdoor air temperature, and (4) average indoor air temperature.

Measuring Wind Velocity. The cup-type of anemometer as used for Weather Bureau observations is sufficiently accurate for this measurement. Some more accurate instruments as well as direct-reading types have been developed for airport service, but for ventilation work it is the average wind velocity over a long period which determines the capacity of the system. Hence the use of the Weather Bureau instrument, with an observation period of one hour or more, is satisfactory. If observations of wind direction are required, these should be taken by observing a sensitive weather vane at frequent intervals (about every 5 minutes) during the same period.

Velocity of Air Through Openings. The vane type anemometer is the most practical instrument for this measurement.

Use a small (4 in.) low-speed anemometer, and correct all readings according to a recent calibration. Mount the anemometer in a strap iron clamp with a long handle for convenience. Divide each opening into 5 in. squares (by string or wire) and hold the anemometer in the center of each square for a definite period of from 15 to 30 seconds. Record the result of the traverse as soon as completed and start another one immediately. A series of traverses over a period of one hour, or the full period covered by the wind velocity observations with a fairly steady wind, may be considered a satisfactory test for that wind velocity. It is preferable to have an anemometer observer at each opening. If the opening is covered by a grille or register, use the proper correction factors (see Chapter 40).

Outdoor Temperature. It is easy to make an error of 1 to 5 deg in observing the outdoor air temperature. An accurate thermometer, calibrated in 1 deg divisions should be used. The thermometer should be mounted in the shade at about mid-height of the building and not too near the building wall or adjacent to an air outlet. The heat from a wall or roof which has been exposed to the sun is easily transmitted to a thermometer, with resulting high readings.

Average Indoor Temperature. It is important to note that the capacity of an opening (such as roof ventilator) does *not* depend on the difference in the temperatures measured adjacent to the opening. It depends rather on the difference between the *average* temperature of the column of air inside the building and that outside. Indoor temperatures should therefore be observed at various heights to secure a good average.

DAIRY BARN VENTILATION[4]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be

[4]For additional information on this subject refer to *Technical Bulletin*, U. S. Department of Agriculture (1930), by M. A. R. Kelley.

Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1933).

supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cfh of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cu ft of barn space, and 0.197 to 0.305 Btu per hour per sq ft of barn exposure.

GARAGE VENTILATION[5, 6]

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the

[5]Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

Airation Study of Garages, by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[6]Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, July, 1933).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, December, 1933).

University of Kansas, Lawrence, Kans., in coöperation with the A.S.H. V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed below:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

PROBLEMS IN PRACTICE

1 ● a. What means are available for the ventilation of buildings?

b. What precaution is necessary in combining different means of ventilating?

a. Natural forces, such as winds and stack effect, and mechanical forces furnished by fans.

b. It is desirable that the different forces used be not in opposition. Their actions should be mutually helpful. For example, a simple roof opening should be placed in the region of lowest pressure caused by a prevailing wind. (See Fig. 1.)

2 ● a. What factors are important in the location and control of ventilating openings?

b. What types of ventilating openings are best suited to a proper distribution of the air supplied?

a. The proper distribution of air as required by the occupants, and the best utilization of natural ventilating forces. The general rules on page 85 apply particularly to these factors.

b. Windows with swinging sash and openings with deflectors may be used to direct air to the points desired.

3 ● a. What is the best location for ventilating openings?

b. How are the sizes of ventilating openings determined for proper air supply?

a. Inlet openings should be low and facing the prevailing winds where possible. Outlet openings should be high and on the side opposite the prevailing winds.
b. For simple openings use Formula 1:

$$Q = EAV$$

and for stacks use Formula 2:

$$Q = 9.4\,A\,\sqrt{H\,(t_1 - t_2)}$$

The use of these formulae is illustrated in Example 1 of the text of this chapter. Inlet and outlet areas should be approximately the same for best results.

4 ● a. What are the advantages of roof ventilators?

b. How are proper sizes determined for roof ventilators?

a. Roof ventilators offer the best utilization of the inductive force of the wind, and they may be very economically fitted with built-in fans to supply the necessary circulation when the force of the wind is not sufficient.

b. Because of the many factors affecting the flow through roof ventilators no accurate formula can be given. It is usual practice to make the combined throat area of all roof ventilators between one-half area and full area of the air inlets as determined by Formula 1.

5 ● What methods of control are used in ventilating systems?

Hand control, control by a thermostat located in the ventilated space or in the ventilator, or wind velocity control designed to keep the air discharge constant regardless of wind velocity.

6 ● How is the quantity of air required for a building determined?

Sufficient air must be supplied to carry away the heat and impurities generated within a building. The temperature rise and concentration of impurities in the exhaust air must be held within specified limits. (See Example 1.)

7 ● What measurements are necessary to determine the capacity of a ventilating system?

Wind velocity and air velocities through openings, determined by suitable cup anemometers; outdoor air temperatures, measured by a shaded thermometer not near objects heated by the sun or near exhaust air openings; indoor air temperatures, measured at various heights to secure a good average.

8 ● How much air must be supplied for dissipating the heat generated in a dairy barn housing 100 cows if the outside temperature is 20 F and the inside temperature is to be maintained at 45 F?

The total heat generated is 100 × 3000 = 300,000 Btu per hour or 5,000 Btu per minute. Then from Formula 3,

$$\begin{aligned} Q &= \frac{HV}{CD} \\ &= \frac{5000 \times 13.5}{0.24 \times (45 - 20)} \\ &= 11{,}250 \text{ cu ft per minute.} \end{aligned}$$

This amount of air should also keep down humidity and odors.

9 ● a. What precaution is necessary in the ventilation of garages using natural ventilation?

b. How much window area is required for a garage with 50 x 100 sq ft floor area if natural ventilation is used?

a. The carbon monoxide content of the air should be kept below 1 part in 10,000 and windows should be kept open at all times.

b. The window area should aggregate 5 per cent of the floor area.

$$0.05 \times 50 \times 100 = 250 \text{ sq ft of window area.}$$

This area should be evenly distributed along two sides of the building.

Chapter 5

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Heat Transfer, Calculations for Transmission Losses, Areas Where Transmission Losses Occur, Coefficients of Transmission, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Constructions

To maintain specified inside temperature conditions and determine the type of plant required, it is essential to know the transmission losses of a structure and consider them in conjunction with the infiltration losses.

Whenever a difference in temperature exists between the two sides of any structural material, such as a wall or roof of a building, a transfer of heat takes place through that material. When the inside temperature is the higher, heat reaches or enters the inside surface of the wall by radiation and convection, because the air and objects within the building are always warmer than the inside surface of the wall when the inside air temperature t is greater than the outside air temperature t_o. This heat must then pass through the material of the wall from the inside to the outside surface by conduction, and is finally given off from the outside surface by radiation and convection, provided, of course, that equilibrium has been established and all four temperatures are constant. If the outside temperature is the higher, the reverse process takes place.

CALCULATIONS FOR TRANSMISSION LOSSES

The calculations for heat transmission losses are made by multiplying the area A in square feet of wall, glass, roof, floor, or material through which the loss takes place, by the proper coefficient U for such construction or material and by the temperature difference between the inside air temperature t at the proper level (in many cases not the *breathing-line*) and the outside air temperature t_o. Therefore,

$$H_t = A\,U\,(t - t_o) \tag{1}$$

where

H_t = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases.)

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

Heat is lost from a building by transmission through all of those surfaces which separate heated spaces from the outside air or from unheated colder spaces within the building. In general, five kinds of surfaces are involved: (1) outside walls; (2) outside glass; (3) inside walls or partitions next to unheated spaces; (4) ceilings of upper floors, either below a cold attic space or as the underside of a roof slab; and (5) floors of heated rooms above an unheated space.

The net inside wall surface is usually determined by reference to the scale plans and elevations of the building concerned. In some cases, of course, the actual building may have to be measured. The total area of all outside openings which are occupied by windows and doors is accurately measured and listed as *glass*. The *glass* area is then deducted from the total outside wall area for each room and the difference is the net wall area. If there are no partitions, measure from the inside face of one wall to the inside face of the next wall. The areas of walls, ceilings and floors next to cold or unheated spaces are found, of course, by taking the inside dimensions of such areas, measured on the heated side.

COEFFICIENTS OF TRANSMISSION

The coefficients of transmission may be determined by means of the guarded hot box or the Nicholls heat meter described in Chapter 40, or they may be calculated from fundamental constants. Because of the unlimited number of combinations of building materials, it would be impractical to attempt to determine by test the heat transmission coefficient of every type of construction in use; consequently, in most cases it is advisable to calculate these coefficients.

Symbols

The following symbols are used in the heat transmission formulae in this chapter:

U = thermal transmittance or over-all coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C_a = thermal conductance per unit area; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plaster board and hollow clay tile.

f = film or surface conductance; the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space; the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of

an air space for a temperature difference of 1 deg F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.:*

$\frac{1}{U}$ = over-all or air-to-air resistance.

$\frac{1}{k}$ = internal resistivity.

$\frac{1}{C_a}$ = internal resistance.

$\frac{1}{f}$ = film or surface resistance.

$\frac{1}{a}$ = air-space resistance.

Fundamental Formulae

The formula of the over-all coefficient for a simple wall x inches thick is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o}} \tag{2}$$

and for a compound wall of several materials having thicknesses in inches of x_1, x_2, x_3, etc., the coefficient is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} + etc.} \tag{3}$$

In the case of air-space construction, an air-space coefficient for each air space must be inserted in either Equation 2 or 3. Thus for a simple wall with one air space,

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o}} \tag{4}$$

and for a simple wall of several air spaces having conductances of a_1, a_2, a_3, etc., the coefficient is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a_1} + \frac{x_2}{k_2} + \frac{1}{a_2} + \frac{x_3}{k_3} + \frac{1}{a_3} + \frac{x_4}{k_4} + \frac{1}{f_o} + etc.} \tag{5}$$

With certain special forms of materials which have irregular air spaces (such as hollow tile) or are otherwise non-homogeneous, it is necessary to use the conductance (C_a) for the unit construction, in which case $\frac{x}{k}$ is replaced by $\frac{1}{C_a}$.

As in the case of the simple wall, f_i and f_o are always the inside and outside surface coefficients for the two materials in contact with air. If

the air is still (no wind), then for the same material f_i and f_o are the same, and $f_i = f_o$; but if the outside air is in motion, then f_o is always greater than f_i and will increase as the wind velocity increases. Values for f_i in still and moving air have been determined for various building materials at the University of Minnesota under a coöperative research agreement with the Society[1]. The range of values for ordinary building materials is comparatively small and for practical purposes may be assumed constant for either still air or any given wind velocity, particularly in view of the fact that the surface resistances usually comprise only a small part of the total resistance of the construction, except in the case of thin, highly conductive walls. In determining basic heat transmission values for building construction, it is customary to use that value of f_o which will occur when a 15-mph wind blows parallel to the outer surfaces considered.

TABLE 1. CONDUCTANCES OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

Mean Temp Deg Fahr	Conductances of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a]Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

The conductances of air spaces at various mean temperatures and widths, for ordinary building materials, are given in Table 1. These results were likewise obtained at the University of Minnesota under a co-operative research agreement with the Society.

Values for k and C_a, the conductivity and conductance of building materials and insulations, are given in Table 2 as taken from the published values of various investigators. It should be noted that values of k and C_a as well as of U are dependent on the mean temperature, and it is therefore desirable that the investigator determine heat-transmission values under conditions approximating those existing under actual conditions. Recommended values for calculating the coefficients of transmission of various types of construction are marked by an asterisk in Table 2.

[1]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930). See also references at end of chapter.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C_a) OF BUILDING MATERIALS AND INSULATORS[a]

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C_a)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance (R)	Authority
MASONRY MATERIALS						
Brick	Common	....	....	5.00*	0.20	....
	Face	....	....	9.20*	0.11	....
Brickwork	Damp or wet	....	....	5.00[b]	0.20	(2)
Cement Mortar	Typical	....	....	12.00*	0.08	....
Cinder Concrete	Typical	110.0	75	5.20*	0.19	(3)
Cinder Blocks[c]	Typical (8 in.)	....	....	0.62†*	1.61	....
	" (12 in.)	....	....	0.51†*	1.96	....
Concrete	Typical	....	....	12.00*	0.08	....
	1-2-4 mix	143.0	69	9.46	0.11	(4)
	Various ages and mixes[d]	....	....	11.35 *to* 16.36	....	(5)
	Cellular	40.0	75	1.06	0.94	(3)
	"	50.0	75	1.44	0.69	(3)
	"	60.0	75	1.80	0.56	(3)
	"	70.0	75	2.18	0.46	(3)
	Typical gypsum fiber concrete, 87.5% gypsum and 12.5% wood chips	51.2	74	1.66*	0.60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101.0	70	3.98	0.25	(3)
Concrete Blocks[c]	Typical (8 in.)	....	....	1.00†*	1.00	....
	" (12 in.)	....	....	0.80†*	1.25	....
	Special concrete block made with an aggregate of hardened clay—4 x 8 x 16 in., 3 cores, 18% voids	74.0	....	0.66†	1.51	(3)
	Special concrete block made with an aggregate of hardened clay—8 x 8 x 16 in., 4 cores, 35% voids	74.5	....	0.30†	3.33	(3)
Stone	Typical	....	....	12.50*	0.08	....
Stucco	"	....	....	12.00*	0.08	....
Tile	Typical hollow clay (4 in.)	....	....	1.00†*	1.00	....
	" " " (6 in.)[e]	....	....	0.64†*	1.57	....
	" " " (8 in.)[e]	....	....	0.60†*	1.67	....
	" " " (10 in.)[e]	....	....	0.58†*	1.72	....
	" " " (12 in.)[e]	....	....	0.40†*	2.50	....
	" " " (16 in.)[e]	....	....	0.31†*	3.23	....
	Hollow clay (2 in.) ½-in. plaster both sides	120.0	110	1.00†	1.00	(2)
	Hollow clay (4 in.) ½-in. plaster both sides	127.0	100	0.60†	1.67	(2)
	Hollow clay (6 in.) ½-in. plaster both sides	124.3	105	0.47†	2.13	(2)
	Hollow gypsum (4 in.)	....	....	0.46†*	2.18	....
	Solid gypsum	51.8	70	1.66	0.60	(4)
	Solid gypsum	75.6	76	2.96	0.34	(4)
Tile or Terrazzo	Typical flooring	....	....	12.00*	0.08	....

AUTHORITIES:

[1]U. S. Bureau of Standards, tests based on samples submitted by manufacturers.
[2]A. C. Willard, L. C. Lichty, and L. A. Harding, tests conducted at the University of Illinois.
[3]J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.
[4]F. B. Rowley, tests conducted at the University of Minnesota.
[5]A.S.H.V.E. Research Laboratory.
[6]E. A. Allcut, tests conducted at the University of Toronto.
[7]Lees and Chorlton.

*Recommended conductivities and conductances for computing heat transmission coefficients.

†For thickness stated or used on construction, not per 1-in. thickness.

[a]For additional conductivity data see Table 14, Page 63, *1934 A.S.R.E. Data Book*.

[b]Recommended value. See *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.

[c]One air cell in the direction of heat flow.

[d]See A.S.H.V.E. Research Paper, *Conductivity of Concrete*, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

[e]The 6-in., 8-in., and 10-in. hollow tile figures are based on two cells in the direction of heat flow. The 12-in. hollow tile is based on three cells in the direction of heat flow. The 16-in. hollow tile consists of one 10-in. and one 6-in. tile, each having two cells in the direction of heat flow.

[f]Not compressed.

[g]Roofing, 0.15-in. thick (1.34 lb per sq ft), covered with gravel (0.83 lb per sq ft), combined thickness assumed 0.25.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C_a) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C_a)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance (R)	Authority
INSULATION—BLANKET OR FLEXIBLE TYPES						
FIBER	Typical	......	....	0.27*	3.70	....
	Chemically treated wood fibers held between layers of strong paper^f	3.62	70	0.25	4.00	(3)
	Eel grass between strong paper^f	4.60	90	0.26	3.85	(1)
	" " " " "	3.40	90	0.25	4.00	(1)
	Flax fibers between strong paper^f	4.90	90	0.28	3.57	(1)
	Hair felt between layers of paper^f	11.00	75	0.25	4.00	(3)
	Kapok between burlap or paper^f	1.00	90	0.24	4.17	(1)
INSULATION—SEMI-RIGID TYPE						
FIBER	Felted cattle hair^f	13.00	90	0.26	3.84	(1)
	" " "	11.00	90	0.26	3.84	(1)
	Flax^f	12.10	70	0.30	3.33	(3)
	Flax and rye^f	13.60	90	0.32	3.12	(1)
	Felted hair and asbestos^f	7.80	90	0.28	3.57	(1)
	75% hair and 25% jute^f	6.30	90	0.27	3.70	(1)
	50% hair and 50% jute^f	6.10	90	0.26	3.85	(1)
	Jute^f	6.70	75	0.25	4.00	(3)
	Felted jute and asbestos^f	10.00	90	0.37	2.70	(1)
	Compressed peat moss	11.00	70	0.26	3.84	(3)
INSULATION—LOOSE FILL OR BAT TYPE						
FIBER	Made from ceiba fibers^f	1.90	75	0.23	4.35	(3)
	" " " "	1.60	75	0.24	4.17	(3)
	Fibrous material made from dolomite and silica	1.50	75	0.27	3.70	(3)
	Fibrous material made from slag	9.40	103	0.27	3.70	(1)
GLASS WOOL	Fibrous material 25 to 30 microns in diameter, made from virgin bottle glass	1.50	75	0.27	3.70	(3)
GRANULAR	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
GYPSUM	Cellular, dry	30.00	90	1.00	1.00	(1)
	" "	24.00	90	0.77	1.30	(1)
	" "	18.00	90	0.59	1.69	(1)
	" "	12.00	90	0.44	2.27	(1)
	Flaked, dry and fluffy^f	34.00	90	0.60	1.67	(1)
	" " " "	26.00	90	0.52	1.92	(1)
	" " " "	24.00	75	0.48*	2.08	(3)
	" " " "	19.80	90	0.35	2.86	(1)
	" " " "	18.00	75	0.34	2.94	(3)
MINERAL WOOL	All forms, typical	......	....	0.27*	3.70	—
REGRANULATED CORK	About 3/16-in. particles	8.10	90	0.31	3.22	(1)
ROCK WOOL	Fibrous material made from rock	21.00	90	0.30	3.33	(1)
	" " " " "	18.00	90	0.29	3.45	(1)
	" " " " "	14.00	90	0.28	3.57	(1)
	" " " " "	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(1)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
SAWDUST	Ordinary	......	86	1.04	0.96	(1)
SHAVINGS	Ordinary	......	86	0.71	1.41	(1)
INSULATION—RIGID						
CORKBOARD	Typical	......	....	0.30*	3.33	....
	No added binder	14.00	90	0.34	2.94	(1)
	" " "	10.60	90	0.30	3.33	(1)
	" " "	7.00	90	0.27	3.70	(1)
	" " "	5.40	90	0.25	4.00	(1)
	Asphaltic binder	14.50	90	0.32	3.12	(1)
FIBER	Typical	......	....	0.33*	3.03	....
	Made from chemically treated wood fiber	20.00	70	0.36	2.78	(3)
	Made from chemically treated wood and vegetable fibers	25.00	75	0.38	2.63	(3)

For notes see Page 95.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C_a) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C_a)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance (R)	Authority
INSULATION—RIGID —Continued						
FIBER	Made from corn stalks	15.00	71	0.33	3.03	(3)
	" " exploded wood fiber	17.90	78	0.32	3.12	(4)
	" " hard wood fibers	15.20	70	0.32	3.12	(3)
	Insulating plaster 9/10-in. thick applied to ⅜-in. plaster board base	54.00	75	1.07†	0.93	(3)
	Made from licorice roots	16.10	81	0.34	2.94	(3)
	Made from 85% magnesia and 15% asbestos	19.30	86	0.51	1.96	(1)
	Made from shredded wood and cement	24.20	72	0.46	2.17	(3)
	" " sugar cane fiber	13.50	70	0.33	3.03	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.80	70	0.30	3.33	(3)
	Made from wheat straw	17.00	68	0.33	3.03	(3)
	" " wood fiber	15.90	72	0.33	3.03	(3)
	" " " "	15.00	70	0.33	3.03	(3)
	" " " "	......	52	0.33	3.03	(6)
	" " " "	8.50	72	0.29	3.45	(3)
	" " " "	15.20	....	0.33	3.03	(3)
	" " " "	16.90	90	0.34	2.94	(1)
BUILDING BOARDS						
ASBESTOS	Compressed cement and asbestos sheets	123.00	86	2.70	0.37	(1)
	Corrugated asbestos board	20.40	110	0.48	2.08	(2)
	Pressed asbestos mill board	60.50	86	0.84	1.19	(1)
	Sheet asbestos	48.30	110	0.29	3.45	(2)
GYPSUM	Gypsum between layers of heavy paper	62.80	70	1.41	0.71	(3)
	Rigid, gypsum between layers of heavy paper (½-in. thick)	53.50	90	2.60†	0.38	(1)
	Gypsum mixed with sawdust between layers of heavy paper (0.39-in. thick)	60.70	90	3.60†	0.28	(1)
PLASTER BOARD	(⅜ in.)	......	....	3.73†*	0.27	....
	(½ in.)	......	....	2.82†*	0.35	....
ROOFING CONSTRUCTION						
ROOFING	Asphalt, composition or prepared	70.00	75	6.50†*	0.15	(3)
	Built up—⅜-in. thick	......	....	3.53†*	0.28	....
	Built up, bitumen and felt, gravel or slag surfaced[g]	......	....	1.33†	0.75	(2)
	Plaster board, gypsum fiber concrete and 3-ply roof covering	52.40	76	0.58†	1.72	(4)
SHINGLES	Asbestos	65.00	75	6.00†*	0.17	(3)
	Asphalt	70.00	75	6.50†*	0.15	(3)
	Slate	201.00	....	10.37*	0.10	(7)
	Wood	......	....	1.28†*	0.78	....
PLASTERING MATERIALS						
PLASTER	Cement	......	....	8.00	0.13	(2)
	Gypsum, typical	......	....	3.30*	0.30	....
	Thickness ⅜ in.	......	73	8.80†	0.11	(4)
METAL LATH AND PLASTER	Total thickness ¾ in.	......	....	4.40†*	0.23	....
WOOD LATH AND PLASTER	⅜-in. plaster, total thickness ¾ in.	......	70	2.50†*	0.40	(4)
BUILDING CONSTRUCTIONS						
FRAME	1-in. fir sheathing and building paper	......	30	0.71†*	1.41	(4)
	1-in. fir sheathing, building paper, and yellow pine lap siding	......	20	0.50†*	2.00	(4)
	1-in. fir sheathing, building paper and stucco	......	20	0.82†*	1.22	(4)
	Pine lap siding and building paper—siding 4 in. wide	......	16	0.85†*	1.18	(4)
	Yellow pine lap siding	......	....	1.28†*	0.78	....
FLOORING	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (¼-in.)	......	....	1.36†*	0.74	....

For notes see Page 95.

Table 2. Conductivities (k) and Conductances (C_a) of Building Materials and Insulators—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C_a)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance (R)	Authority
AIR SPACE AND SURFACE COEFFICIENTS						
Air Spaces	Over ¾-in. faced with ordinary building materials	....	40	1.10†*	0.91	(4)
Surfaces, Ordinary	Still air (f_i)	....	....	1.65†*	0.61	(4)
	15 mph—(f_o)	....	....	6.00†*	0.17	(4)
Surface, Bright Aluminum	Still air (f_i)	....	60	1.18†	0.85	....
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL						
	Air space, faced one side with bright aluminum foil, over ¾-in. wide	....	50	0.46†*	2.17	(4)
	Air space, faced one side with bright aluminum foil, ⅜-in. wide	....	50	0.62†	1.61	(4)
	Air space, faced both sides with bright aluminum foil, over ¾-in. wide	....	50	0.41†*	2.44	(4)
	Air space, faced both sides with bright aluminum foil, ⅜-in. wide	....	50	0.57†	1.75	(4)
	Air space divided in two with single curtain of bright aluminum foil (both sides bright)					
	Each space over ¾-in. wide	....	50	0.23†*	4.35	(4)
	Each space ⅜-in. wide	....	50	0.31†	3.23	(4)
	Air space with multiple curtains of bright aluminum foil, bright on both sides, curtains more than ¾-in. apart, in standard construction					
	2 curtains, forming 3 spaces	....	50	0.15†*	6.67	(4)
	3 curtains, forming 4 spaces	....	50	0.11†*	9.09	(4)
	4 curtains, forming 5 spaces	....	50	0.09†*	11.11	(4)
WOODS (Across Grain)						
Balsa	....	20.0	90	0.58	1.72	(1)
	....	8.8	90	0.38	2.63	(1)
	....	7.3	90	0.33	3.03	(1)
California Redwood	0% moisture	22.0	75	0.66	1.53	(4)
	0% "	28.0	75	0.70	1.43	(4)
	8% "	22.0	75	0.70	1.43	(4)
	8% "	28.0	75	0.75	1.33	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	28.0	75	0.80	1.25	(4)
Cypress	....	28.7	86	0.67	1.49	(1)
Douglas Fir	0% moisture	26.0	75	0.61	1.64	(4)
	0% "	34.0	75	0.67	1.49	(4)
	8% "	26.0	75	0.66	1.52	(4)
	8% "	34.0	75	0.75	1.33	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
Eastern Hemlock	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	8% "	22.0	75	0.63	1.59	(4)
	8% "	30.0	75	0.81	1.23	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)
Hard Maple	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	8% "	40.0	75	1.08	0.93	(4)
	8% "	46.0	75	1.13	0.89	(4)
	16% "	40.0	75	1.15	0.87	(4)
	16% "	46.0	75	1.21	0.83	(4)

For notes see Page 95.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C_a) OF BUILDING MATERIALS AND INSULATORS—Continued

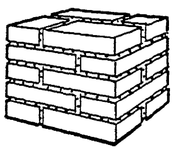
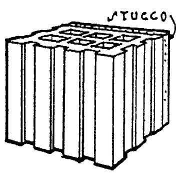
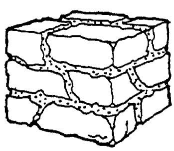
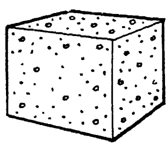
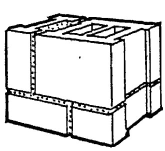
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	Mean Temp. (Deg Fahr)	Conductivity (k) or Conductance (C_a)	Resistivity $\left(\frac{1}{k}\right)$ or Resistance (R)	Authority
WOODS—Continued						
Longleaf Yellow Pine	0% moisture	30.0	75	0.76	1.32	(4)
	0% "	40.0	75	0.86	1.16	(4)
	8% "	30.0	75	0.83	1.21	(4)
	8% "	40.0	75	0.95	1.05	(4)
	16% "	30.0	75	0.89	1.12	(4)
	16% "	40.0	75	1.03	0.97	(4)
Mahogany		34.3	86	0.90	1.11	(1)
Maple		44.3	86	1.10	0.91	(1)
Maple or Oak				1.15*	0.87	
Norway Pine	0% moisture	22.0	75	0.62	1.61	(4)
	0% "	32.0	75	0.74	1.35	(4)
	8% "	22.0	75	0.68	1.47	(4)
	8% "	32.0	75	0.83	1.21	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.91	1.10	(4)
Red Cypress	0% moisture	22.0	75	0.67	1.49	(4)
	0% "	32.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.71	1.41	(4)
	8% "	32.0	75	0.84	1.19	(4)
	16% "	22.0	75	0.74	1.35	(4)
	16% "	32.0	75	0.90	1.11	(4)
Red Oak	0% moisture	38.0	75	0.98	1.02	(4)
	0% "	48.0	75	1.18	0.85	(4)
	8% "	38.0	75	1.03	0.97	(4)
	8% "	48.0	75	1.24	0.81	(4)
	16% "	38.0	75	1.07	0.94	(4)
	16% "	48.0	75	1.29	0.78	(4)
Shortleaf Yellow Pine	0% moisture	26.0	75	0.74	1.35	(4)
	0% "	36.0	75	0.91	1.10	(4)
	8% "	26.0	75	0.79	1.27	(4)
	8% "	36.0	75	0.97	1.03	(4)
	16% "	26.0	75	0.84	1.19	(4)
	16% "	36.0	75	1.04	0.96	(4)
Soft Elm	0% moisture	28.0	75	0.73	1.37	(4)
	0% "	34.0	75	0.88	1.14	(4)
	8% "	28.0	75	0.77	1.30	(4)
	8% "	34.0	75	0.93	1.08	(4)
	16% "	28.0	75	0.81	1.24	(4)
	16% "	34.0	75	0.97	1.03	(4)
Soft Maple	0% moisture	36.0	75	0.89	1.12	(4)
	0% "	42.0	75	0.95	1.05	(4)
	8% "	36.0	75	0.96	1.04	(4)
	8% "	42.0	75	1.02	0.98	(4)
	16% "	36.0	75	1.01	0.99	(4)
	16% "	42.0	75	1.09	0.92	(4)
Sugar Pine	0% moisture	22.0	75	0.54	1.85	(4)
	0% "	28.0	75	0.64	1.56	(4)
	8% "	22.0	75	0.59	1.70	(4)
	8% "	28.0	75	0.71	1.41	(4)
	16% "	22.0	75	0.65	1.54	(4)
	16% "	28.0	75	0.78	1.28	(4)
Virginia Pine		34.3	86	0.96	1.04	(1)
West Coast Hemlock	0% moisture	22.0	75	0.68	1.47	(4)
	0% "	30.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.73	1.37	(4)
	8% "	30.0	75	0.85	1.18	(4)
	16% "	22.0	75	0.78	1.28	(4)
	16% "	30.0	75	0.91	1.10	(4)
White Pine		31.2	86	0.78	1.28	(1)
Yellow Pine				1.00	1.00	(3)
Yellow Pine or Fir				0.80*	1.25	

For notes see Page 95.

TABLE 3. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL	THICKNESS OF MASONRY (INCHES)	WALL No.
	**Solid Brick** Based on 4-in. face brick and the remainder common brick.	8 12 16	**1** **2** **3**
	**Hollow Tile** Stucco Exterior Finish. The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in. tile each having two cells in the direction of heat flow.	8 10 12 16	**4** **5** **6** **7**
	**Limestone or Sandstone**	8 12 16 24	**8** **9** **10** **11**
	**Concrete** These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.	6 10 16 20	**12** **13** **14** **15**
	**Hollow Cinder Blocks** Based on one air cell in direction of heat flow.	8 12	**16** **17**
	Hollow Concrete Blocks Based on one air cell in direction of heat flow.	8 12	**18** **19**

[a]Computed from factors marked by * in Table 2.
[b]Based on the actual thickness of 2-in. furring strips.

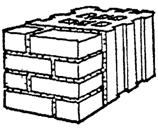
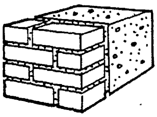
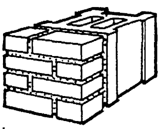
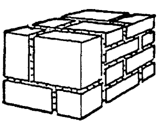
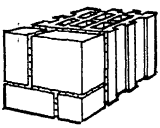
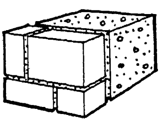
INTERIOR FINISH

Uninsulated Walls						Insulated Walls					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	Decorated building board (½ in.) without plaster—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster (¾ in.) on metal lath attached to furring strips—furred space (over ¾-in. wide) faced one side with bright aluminum foil	Plaster on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[c]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.23	0.12	0.20
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.14	0.12	0.19	0.11	0.17
0.28	0.27	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.40	0.37	0.26	0.27	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.30	0.29	0.22	0.22	0.22	0.17	0.17	0.13	0.12	0.17	0.10	0.16
0.25	0.24	0.19	0.19	0.19	0.15	0.15	0.12	0.11	0.15	0.097	0.14
0.71	0.64	0.37	0.39	0.37	0.26	0.25	0.18	0.15	0.26	0.13	0.23
0.58	0.53	0.33	0.34	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.49	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.79	0.70	0.39	0.42	0.39	0.27	0.26	0.19	0.16	0.27	0.13	0.23
0.62	0.57	0.34	0.37	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.48	0.44	0.29	0.31	0.29	0.22	0.21	0.16	0.14	0.22	0.12	0.20
0.41	0.39	0.27	0.28	0.27	0.21	0.20	0.15	0.13	0.21	0.12	0.18
0.42	0.39	0.27	0.28	0.27	0.21	0.20	0.16	0.13	0.21	0.12	0.19
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.56	0.52	0.32	0.34	0.32	0.24	0.23	0.17	0.14	0.24	0.12	0.21
0.49	0.46	0.30	0.32	0.30	0.23	0.22	0.16	0.14	0.23	0.12	0.20

[c]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS WITH VARIOUS TYPES OF VENEERS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL		WALL No.
	FACING	BACKING	
	4 in. Brick Veneer[d]	6 in. 8 in. 10 in. 12 in. Hollow Tile[c]	20 21 22 23
	4 in. Brick Veneer[d]	6 in. 10 in. 16 in. Concrete	24 25 26
	4 in. Brick Veneer[d]	8 in. 12 in. Cinder Blocks[e]	27 28
	4 in. Brick Veneer[d]	8 in. 12 in. Concrete Blocks[e]	29 30
	4 in. Cut-Stone Veneer[d]	8 in. 12 in. 16 in. Common Brick	31 32 33
	4 in. Cut-Stone Veneer[d]	6 in. 8 in. 10 in. 12 in. Hollow Tile[c]	34 35 36 37
	4 in. Cut-Stone Veneer[d]	6 in. 10 in. 16 in. Concrete	38 39 40

[a]Computed from factors marked by * in Table 2.
[b]Based on the actual thickness of 2-in. furring strips.
[c]The 6-in., 8-in. and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile is based on three cells in the direction of heat flow.

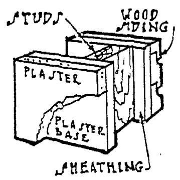
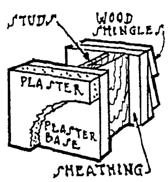
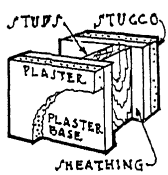
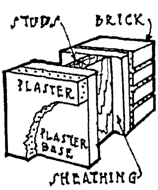
INTERIOR FINISH

Uninsulated Walls					Insulated Walls						
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (¾ in.) on metal lath—furred	Plaster (½ in.) on plaster board (⅜ in.)—furred	No plaster—decorated rigid or building board interior finish (½ in.)—furred	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster on metal lath (¾ in.) attached to furring strips—furred space (over ¾-in. wide) faced one side with bright aluminum foil	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b])[f]	Plaster (¾ in.) on metal lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.16	0.13	0.19	0.11	0.17
0.34	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.34	0.32	0.23	0.24	0.23	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.27	0.26	0.20	0.21	0.20	0.16	0.16	0.13	0.11	0.16	0.10	0.15
0.57	0.53	0.33	0.35	0.33	0.24	0.23	0.17	0.14	0.24	0.13	0.21
0.48	0.45	0.30	0.31	0.30	0.22	0.22	0.16	0.14	0.22	0.12	0.20
0.39	0.37	0.26	0.27	0.26	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.31	0.30	0.22	0.23	0.22	0.18	0.17	0.14	0.12	0.18	0.11	0.16
0.44	0.42	0.28	0.30	0.28	0.21	0.21	0.16	0.13	0.21	0.12	0.19
0.40	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.20	0.11	0.18
0.37	0.35	0.25	0.26	0.25	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.28	0.27	0.21	0.21	0.21	0.17	0.16	0.13	0.12	0.17	0.10	0.15
0.23	0.22	0.18	0.18	0.18	0.15	0.14	0.12	0.11	0.15	0.095	0.14
0.37	0.35	0.25	0.26	0.25	0.20	0.19	0.15	0.13	0.20	0.11	0.18
0.36	0.34	0.24	0.25	0.24	0.19	0.19	0.15	0.13	0.19	0.11	0.17
0.35	0.33	0.24	0.25	0.24	0.19	0.18	0.14	0.12	0.19	0.11	0.17
0.28	0.26	0.20	0.21	0.20	0.17	0.16	0.13	0.11	0.17	0.10	0.15
0.61	0.56	0.34	0.36	0.34	0.25	0.24	0.18	0.15	0.25	0.13	0.22
0.51	0.47	0.31	0.32	0.31	0.23	0.22	0.17	0.14	0.23	0.12	0.20
0.41	0.38	0.26	0.28	0.26	0.20	0.20	0.15	0.13	0.21	0.11	0.18

[d]Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e]Based on one air cell in direction of heat flow.
[f]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 5. COEFFICIENTS OF TRANSMISSION (*U*) OF VARIOUS TYPES OF FRAME CONSTRUCTION[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	EXTERIOR FINISH	TYPE OF SHEATHING	WALL No.
	Wood Siding or Clapboard	1 in. Wood[d]	41
		½ in. Rigid Insulation	42
		½ in. Plaster Board	43
	Wood Shingles	1 in. Wood[d]	44
		½ in. Rigid Insulation[e]	45
		½ in. Plaster Board[e]	46
	Stucco	1 in. Wood[d]	47
		½ in. Rigid Insulation	48
		½ in. Plaster Board	49
	Brick[f] Veneer	1 in. Wood[d]	50
		½ in. Rigid Insulation	51
		½ in. Plaster Board	52

[a]Computed from factors marked by * in Table 2.

[b]These coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plaster board.

[c]Based on the actual width of 2 by 4 studding, namely, 3⅝ in.

INTERIOR FINISH

No Insulation Between Studding							Insulation Between Studding		
Plaster on wood lath on studding	Plaster (¾ in.) on metal lath on studding	Plaster (½ in.) on plaster board (⅜ in.) on studding	Plaster (½ in.) on rigid insulation (½ in.) on studding	Plaster (½ in.) on rigid insulation (1 in.) on studding	Plaster (½ in.) on corkboard (1½ in.) on studding	No plaster—decorated rigid or building board interior finish (½ in.)	Plaster (¾ in.) on metal lath—stud space faced one side with bright aluminum foil	Plaster (¾ in.) on metal lath[b] on studding—rock wool fill (3⅝ in.[c]) between studding[g]	Plaster (¾ in.) on metal lath[b] on studding—flexible insulation (½ in.) between studding and in contact with sheathing
A	B	C	D	E	F	G	H	I	J
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.19	0.061	0.17
0.23	0.24	0.23	0.18	0.14	0.11	0.18	0.18	0.060	0.17
0.31	0.33	0.31	0.22	0.17	0.13	0.23	0.23	0.064	0.20
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.19	0.061	0.17
0.19	0.20	0.19	0.15	0.12	0.10	0.16	0.16	0.057	0.14
0.24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.061	0.17
0.30	0.31	0.30	0.22	0.16	0.12	0.22	0.22	0.064	0.20
0.27	0.29	0.27	0.20	0.16	0.12	0.21	0.21	0.062	0.19
0.40	0.43	0.40	0.26	0.19	0.14	0.28	0.28	0.067	0.24
0.27	0.28	0.27	0.20	0.15	0.12	0.21	0.21	0.062	0.18
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.20	0.061	0.18
0.35	0.37	0.35	0.24	0.18	0.13	0.25	0.25	0.066	0.22

[d]Yellow pine or fir—actual thickness about 25/32 in.

[e]Furring strips between wood shingles and sheathing.

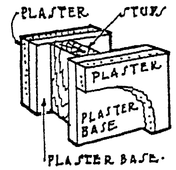
[f]Small air space and mortar between building paper and brick veneer neglected.

[g]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) OF FRAME INTERIOR WALLS AND PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



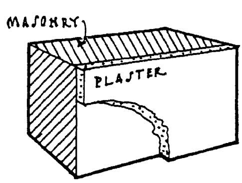
TYPICAL CONSTRUCTION	WALL No.	SINGLE PARTITION (FINISH ON ONE SIDE OF STUDDING)	DOUBLE PARTITION (FINISHED ON BOTH SIDES OF STUDDING)				
			Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½-in. Flexible Insulation Between Studding (One Air Space)	Stud Space Faced One Side with Bright Aluminum Foil
TYPE OF WALL		A	B	C	D	E	F
Wood Lath and Plaster On Studding	53	0.62	0.34	0.11	0.065	0.21	0.24
Metal Lath and Plaster[c] On Studding	54	0.69	0.39	0.11	0.066	0.23	0.26
Plaster Board (⅜ in.) **and Plaster**[d] On Studding	55	0.61	0.34	0.10	0.065	0.21	0.24
½ in. **Rigid Insulation and Plaster**[d] On Studding	56	0.35	0.18	0.083	0.056	0.14	0.15
1 in. **Rigid Insulation and Plaster**[d] On Studding	57	0.23	0.12	0.066	0.048	0.097	0.10
1½ in. **Corkboard and Plaster**[d] On Studding	58	0.16	0.081	0.052	0.040	0.070	0.073
2 in. **Corkboard and Plaster**[d] On Studding	59	0.12	0.063	0.045	0.035	0.057	0.059

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾-in. thick.
[d]Plaster assumed ½-in. thick.

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION

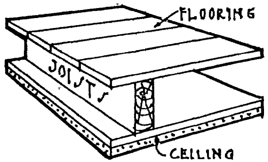


TYPICAL CONSTRUCTION	No.	PLAIN WALLS (NO PLASTER)	WALLS PLASTERED ON ONE SIDE	WALLS PLASTERED ON BOTH SIDES
TYPE OF WALL		A	B	C
4-in. Hollow Clay Tile	**60**	0.45	0.42	0.40
4-in. Common Brick	**61**	0.50	0.46	0.43
4-in. Hollow Gypsum Tile	**62**	0.30	0.28	0.27
2-in. Solid Plaster	**63**	....	....	0.53

[a]Computed from factors marked by * in Table 2.

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.



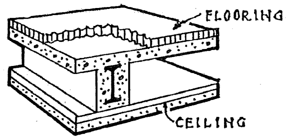
TYPICAL CONSTRUCTION	INSULATION BETWEEN JOISTS	No.	TYPE OF FLOORING				
			No Flooring	Yellow Pine Flooring[b] on Joists	Yellow Pine Flooring on Rigid Insulation (½ in.) on Joists	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring[b] on Joists	¼-in. Battleship Linoleum on Yellow Pine Flooring[b]
TYPE OF CEILING			A	B	C	D	E
No Ceiling	None	1	....	0.46	0.27	0.34	0.34
Metal Lath and Plaster (¾ in.)	None	2	0.69	0.30	0.21	0.25	0.25
Wood Lath and Plaster	None	3	0.62	0.28	0.20	0.24	0.24
Plaster Board (⅜ in.) and Plaster (½ in.)	None	4	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (½ in.) and Plaster (½ in.)	None	5	0.35	0.21	0.16	0.18	0.18
Metal Lath and Plaster	Flexible[d] Insulation (½ in.)	6	0.24	0.16	0.13	0.15	0.15
Metal Lath and Plaster	Rigid Insulation[d] (½ in.)	7	0.26	0.17	0.14	0.15	0.15
Metal Lath and Plaster	Bright Aluminum Foil[e]	8	0.59	0.22	0.16	0.19	0.19
Metal Lath and Plaster	Rock Wool Fill (3⅝ in.)	9	0.067	0.063	0.058	0.060	0.060
Corkboard (1½ in.) and Plaster (½ in.)	None	10	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and Plaster (½ in.)	None	11	0.12	0.10	0.087	0.094	0.094

[a]Computed from factors marked by * in Table 2.
[b]Thickness assumed to be 25/32 in.
[c]Thickness assumed to be 13/16 in.
[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in. furring strips.
[e]Air space faced on one side with bright aluminum foil.

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION



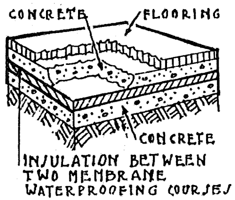
Type of Ceiling	Thickness of Concrete (Inches)	No.	TYPE OF FLOORING: No Flooring (Concrete Bare)[b]	Yellow Pine Flooring[c] on Wood Sleepers Embedded in Concrete[d]	Maple or Oak Flooring[e] on Yellow Pine Sub-Flooring[c] on Wood Sleepers Embedded in Concrete	Tile or Terrazzo[f] Flooring on Concrete	¼-in. Battleship Linoleum Directly on Concrete
			A	B	C	D	E
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44
	6	2	0.59	0.37	0.30	0.56	0.41
	8	3	0.53	0.35	0.28	0.51	0.38
	10	4	0.49	0.33	0.27	0.47	0.36
½ in. **Plaster Applied Directly to Under Side of Concrete**	4	5	0.59	0.38	0.30	0.56	0.41
	6	6	0.54	0.35	0.28	0.52	0.38
	8	7	0.50	0.33	0.27	0.47	0.36
	10	8	0.45	0.32	0.26	0.44	0.34
Suspended or Furred Metal Lath and Plaster (¾ in.) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29
	6	10	0.35	0.26	0.22	0.34	0.28
	8	11	0.33	0.25	0.21	0.32	0.27
	10	12	0.32	0.24	0.21	0.31	0.25
Suspended or Furred Ceiling of Plaster Board (⅜ in.) and Plaster (½ in.)	4	13	0.35	0.26	0.22	0.34	0.28
	6	14	0.33	0.25	0.21	0.32	0.26
	8	15	0.31	0.24	0.21	0.30	0.25
	10	16	0.30	0.23	0.20	0.29	0.24
Suspended or Furred Ceiling of Rigid Insulation (½ in.) and Plaster (½ in.)	4	17	0.24	0.20	0.17	0.24	0.21
	6	18	0.23	0.19	0.17	0.23	0.20
	8	19	0.22	0.18	0.16	0.22	0.19
	10	20	0.22	0.18	0.16	0.21	0.19
Plaster (½ in.) on Corkboard (1½ in.) Set in Cement Mortar (½ in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14
	6	22	0.14	0.13	0.12	0.14	0.13
	8	23	0.14	0.12	0.11	0.14	0.13
	10	24	0.14	0.12	0.11	0.14	0.13

[a]Computed from factors marked by * in Table 2.
[b]The figures in COLUMN A may be used with sufficient accuracy for concrete floors covered with carpet.
[c]Thickness of yellow pine flooring assumed to be 25/32 in.
[d]The figures in COLUMN B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers.
[e]Thickness of maple or oak flooring assumed to be 13/16 in.
[f]Thickness of tile or terrazzo assumed 1 in.

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a, e]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions.

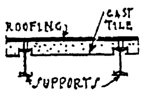
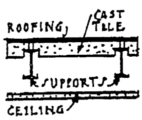
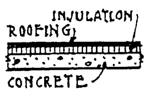
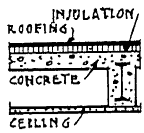
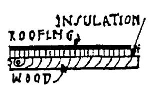
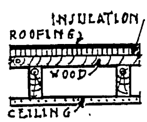
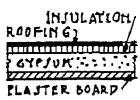
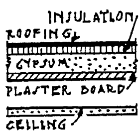
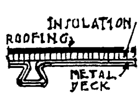
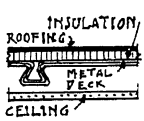
TYPICAL CONSTRUCTION



TYPICAL CONSTRUCTION	THICKNESS OF CONCRETE (INCHES)	No.	TYPE OF FINISH FLOORING				
			No Flooring (Concrete Bare)	Yellow Pine Flooring[b] on Wood Sleepers Resting on Concrete	Maple or Oak Flooring[c] on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	Tile or Terrazzo[d] on Concrete	¼-in. Battleship Linoleum Directly on Concrete
TYPE AND THICKNESS OF INSULATION			A	B	C	D	E
None	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None[e]	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
1 in. **Rigid Insulation**[e]	4	7	0.22	0.16	0.14	0.22	0.19
1 in. **Rigid Insulation**[e]	8	8	0.21	0.15	0.13	0.20	0.18
2 in. **Corkboard**[e]	4	9	0.12	0.099	0.093	0.12	0.11
2 in. **Corkboard**[e]	8	10	0.12	0.096	0.090	0.12	0.11

[a]Computed from factors marked by * in Table 2.
[b]Assumed 25/32 in. thick.
[c]Assumed 13/16 in. thick.
[d]Assumed 1 in. thick.
[e]The figures for Nos. 5 to 10, inclusive, include 3-in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

TABLE 11. COEFFICIENTS OF TRANSMISSION (*U*) OF VARIOUS TYPES OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING[a]

TYPICAL CONSTRUCTION		TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	No.
WITHOUT CEILINGS	WITH METAL LATH AND PLASTER CEILINGS[d]			
		**Precast Cement Tile**	1⅝	**1**
		**Concrete** **Concrete** **Concrete**	2 4 6	**2** **3** **4**
		**Wood** **Wood** **Wood** **Wood**	1[b] 1½[b] 2[b] 4[b]	**5** **6** **7** **8**
		**Gypsum Fiber Concrete**[c] (2 in.) **on Plaster Board** (⅜ in.) **Gypsum Fiber Concrete**[c] (3 in.) **on Plaster Board** (⅜ in.)	2⅜ 3⅜	**9** **10**
		**Flat Metal Roofs** Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.	......	**11**

[a]Computed from factors marked by * in Table 2.
[b]Nominal thicknesses specified—actual thicknesses used in calculations.
[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

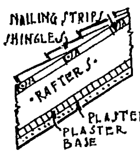
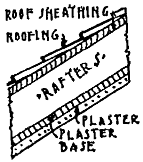
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[d]							
No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[d]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF ROOFING AND ROOF SHEATHING	INSULATION BETWEEN ROOF RAFTERS	No.	TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)								
				No Ceiling (Rafters Exposed)	Metal Lath and Plaster (3/4 in.)	Plaster Board (3/8 in.) and Plaster (1/2 in.)	Wood Lath and Plaster	Rigid Insulation (1/2 in.)	Rigid Insulation (1/2 in.) and Plaster (1/2 in.)	Rigid Insulation (1 in.) and Plaster (1/2 in.)	Corkboard (1½ in.) and Plaster (1/2 in.)	Corkboard (2 in.) and Plaster (1/2 in.)
				A	B	C	D	E	F	G	H	I
	**Wood Shingles on Wood Strips**[b]	**None**	1	0.46	0.30	0.29	0.29	0.22	0.21	0.16	0.12	0.10
		½ in. Flexible[c]	2	....	0.17	0.16	0.16	0.14	0.13	0.11	0.091	0.079
		1 in. Flexible[c]	3	....	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
		Bright Aluminum Foil[g]	4	....	0.22	0.21	0.21	0.17	0.17	0.13	0.10	0.085
		3⅝ in. Rock Wool[e]	5	....	0.063	0.062	0.062	0.058	0.058	0.053	0.048	0.044
	**Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing**[d] **on Wood Sheathing**[f]	**None**	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
		½ in. Flexible[c]	7	....	0.18	0.17	0.17	0.14	0.14	0.12	0.094	0.089
		1 in. Flexible[c]	8	....	0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
		Bright Aluminum Foil[g]	9	....	0.24	0.23	0.23	0.18	0.18	0.14	0.11	0.093
		3⅝ in. Rock Wool[e]	10	....	0.065	0.064	0.064	0.060	0.059	0.054	0.049	0.045

[a]Computed from factors marked by * in Table 2. Nos. 6 to 10, inclusive, based on ½-in. thick slate.
[b]Based on 1 in. by 4 in. strips spaced 2 in.
[c]Figures based on two air spaces. Insulation may also be applied to under side of roof rafters with furring strips between.
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e]Assumed 3⅝ in. thick based on the actual width of 2 in. by 4 in. rafters.
[f]Sheathing assumed 25/32 in. thick.
[g]Air space faced on one side with bright aluminum foil.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS AND SKYLIGHTS

Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window or skylight

A. Windows and Skylights

	U
Single	1.13[a, c]
Double	0.45[a]
Triple	0.281[a]

B. Solid Wood Doors[b, c]

NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U
1	25/32	0.69
1¼	1 1/16	0.59
1½	1 5/16	0.52
1¾	1⅜	0.51
2	1⅝	0.46
2½	2⅛	0.38
3	2⅝	0.33

[a]See *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.
[b]Computed using C = 1.15 for wood; f_i = 1.65 and f_o = 6.0.
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperatures.

While most building materials have surfaces which show similar characteristics as far as the transmission of heat is concerned, it is a well-known fact that certain surfaces such as aluminum bronze, gold bronze, aluminum foil, or in fact any metallic, highly polished surface presents a greater resistance to heat transmission than the surface of the average building material.

The greater heat resistance of such metallic surfaces is due primarily to their higher reflectivity and consequent lower emissivity of radiant heat. The use of multiple layers of metallic surfaces, combined with air spaces of low resistance, provides a definite insulating effect. Factors[2] for air spaces bounded by aluminum foil are given in Table 2.

Coefficients of transmission of various types of wall, ceiling, floor and roof construction with aluminum insulation can be readily calculated. The present installation practice indicates that air spaces of ½ in. to 1½ in. are preferred but manufacturers' recommendations should be closely followed in the application of aluminum foil insulation.

The majority of the conductivities and conductances of the building materials and insulations given in Table 2 were determined by the hot-plate method of testing[3]. Attention is called to the fact that conductivities per inch of thickness of materials or insulations do not afford a true basis for comparison, although they are frequently used for that purpose.

[2]Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1934, p. 263).
[3]Standard Test Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928). See also Chapter 40.

Correct comparisons should take into consideration many different factors, including conductivities or conductances, thicknesses installed and manner of installation, while the selection of an insulation should also give consideration to structural qualities, as well as to material and application costs. Fire, vermin, and rot resistance are other important factors to be considered when comparing materials. At present there is no universally recognized method of rating insulations. Conductivities and conductances of building materials and insulations are useful to the heating engineer in determining over-all coefficients of heat transmission of walls, floors, roofs and ceilings.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of face brick having a conductivity of 9.20, and that the inside course is of common brick having a conductivity of 5.0, the thicknesses each being 4 in. The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively, for still air and a 15 mph wind velocity.

Solution. k (face brick) = 9.20; x = 4.0 in.; k (common brick) = 5.0; x = 4.0 in.; k (plaster) = 3.3; x = ½ in.; f_i = 1.65; f_o = 6.0. Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 2, 3, 4 and 5 and the values of k (or C_a), f_i, f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example, the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}}$$ = 0.48 Btu per hour per square foot per degree difference in temperature between the ground and the air immediately above the floor.

The thicknesses upon which the coefficients in Tables 3 to 13, inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	¾ in.

Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	½ in.
Slate (roofing)	½ in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	⅜ in.
1-in. lumber (S 2-S)	25/32 in.
1½-in. lumber (S-2-S)	1 5/16 in.
2-in lumber (S-2-S)	1⅝ in.
2½-in. lumber (S-2-S)	2⅛ in.
3-in. lumber (S-2-S)	2⅝ in.
4-in. lumber (S-2-S)	3⅝ in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4-in. face brick and the remainder common brick. Stucco is assumed to be 1-in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

Rigid insulation refers to the so-called board form which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top-floor ceiling, unheated attic space, and pitched roof, per square foot of roof area, is as follows:

$$U = \frac{U_r \times U_{ce}}{n \times U_r + U_{ce}} \tag{6}$$

where

U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

In using this formula, a correction factor must be applied. As the amount of heat transferred through an air space is proportional to the difference of the fourth powers of the absolute temperatures of the surfaces enclosing the air space, a greater amount of heat is absorbed or emitted by radiation by the surfaces enclosing an unheated attic than by the surfaces of a wall or ceiling in a room under still-air conditions, where the surrounding objects are only slightly higher in temperature than the interior surfaces of the walls and ceiling. For example, the average

coefficient of a surface in still air is 1.65 Btu per hour per square foot per degree Fahrenheit, whereas the average coefficient of an air space in an outside wall is about 1.10 Btu per hour per square foot per degree Fahrenheit difference between the two surfaces, at a mean temperature of 40 F. An air space coefficient of 1.10 is equivalent to a surface coefficient of 2.20 for each of the two surfaces enclosing the air space, where the over-all transmission is computed by using the coefficients of the two surfaces enclosing the air space instead of the coefficient of the air space itself. Hence, in determining the values of U_r and U_{ce} to be used in the formula, the coefficients for the surfaces of the roof and ceiling enclosing the attic should be increased to allow for the additional amount of heat transferred by radiation, and a coefficient of 2.20 may be used with sufficient accuracy for each of these surfaces, although in very precise work a correction should be made to allow for the fact that the area of a pitched roof over an unheated attic is greater than the area of the ceiling, and hence, the amount of heat absorbed by radiation by each square foot of roof surface is less than is given off by radiation by each square foot of ceiling surface.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall surfaces, the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top-floor ceiling, but it should be noted that these coefficients should be multiplied by the roof area and not by the ceiling area. If the unheated attic contains windows, ventilators or vertical wall surfaces, which would tend to reduce temperature in the attic to a temperature approaching or equaling the outside temperature, the roof should be neglected and only the top-floor ceiling construction and the corresponding ceiling area taken into consideration, using the coefficients given in Tables 8 or 9. Where there are no dormers, doors, or windows, and when the transmission coefficients of the roof and the ceiling are approximately the same, the value of the attic temperature may be taken as an average between the inside and the outside temperature.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms of 32 F.

Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 7.

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Heat Insulation as Applied to Buildings and Structures, by E. A. Allcut, University of Toronto, 1934.

PROBLEMS IN PRACTICE

1 ● What is the conductance of a 1-in. air space, faced with common building materials, at a mean temperature of 50 F?

1.152 (Table 1).

2 ● What is the conductivity of face brick?

9.20 (Table 2).

3 ● What is the conductance of wood shingles?

1.28 (Table 2).

4 ● What is the over-all coefficient of transmission U for a solid brick wall 12-in. thick with plaster on wood lath, furred?

0.24 (Table 3, Wall 2C).

5 ● Find the value of U for a 6-in. concrete wall with plaster on metal lath attached to 2-in. furring strips with flanged ½-in. blanket insulation.

0.23 (Table 3, Wall 12L).

6 ● Find the value of *U* for a wood siding wall with an interior finish of ¾-in. plaster on metal lath; sheathing thickness, 25/32 in.

0.26 (Table 5, Wall 41B).

7 ● What value of *U* should be used for a brick veneer wall with ½-in. rigid insulation sheathing finished on the interior with plaster on ½-in. rigid insulation?

0.19 (Table 5, Wall 51D).

8 ● What value of *U* should be used in computing the heat loss from an attic through a floor of yellow pine on joists with a ceiling of metal lath and plaster?

0.30 (Table 8, Floor 2B).

9 ● What is the over-all heat transfer coefficient for a 6-in. concrete floor with no insulation and with yellow pine flooring on sleepers resting on concrete?

0.33 (Table 10, Floor 2B).

10 ● What is the coefficient *U* for a flat roof of 4-in. concrete with a metal lath and plaster ceiling insulated with 1-in. cork board?

0.17 (Table 11, Roof 3N).

11 ● A solid 12-in. common brick wall is finished on the inside with ½-in. insulation plaster base, and ½-in. plaster; the plaster base is furred 1 in. from the brick; k for insulating material = 0.34. Calculate the over-all coefficient *U*.

$$f_i = 1.65; f_o = 6.00; k \text{ for brick} = 5.00; a \text{ for 1-in. air space} = 1.1$$

$$\text{Over-all heat resistance} = R = \frac{1}{1.65} + \frac{.5}{3.3} + \frac{1}{1.1} + \frac{12}{5} + \frac{1}{6} = 5.703$$

$$U = \frac{1}{R} = 0.175$$

12 ● A wall is built with two layers of ½-in. insulating material spaced 1 in. apart; the air space is lined on one side with bright aluminum foil; mean temperature is 40 F; still air on both sides of wall; *k* for insulating material is 0.34. Calculate the value of *U*.

$$f_i = 1.65; f_o = 1.65; a = 0.46$$

$$R = \frac{1}{1.65} + \frac{0.5}{0.34} + \frac{1}{0.46} + \frac{0.5}{0.34} + \frac{1}{1.65} = 6.327$$

$$U = \frac{1}{R} = 0.158$$

13 ● What is the inside surface temperature of a 6-in. solid concrete wall? Inside air, 70 F; outside air, −20 F with 15 mph wind.

The temperature drop from point to point through a wall is directly proportional to the heat resistance.

$$f_i = 1.65; k \text{ for concrete} = 12; f_o = 6.0$$

$$\text{Over-all resistance } R = \frac{1}{1.65} + \frac{6}{12} + \frac{1}{6.0} = 1.27$$

$$\frac{\text{Temperature drop, inside air to surface}}{\text{Temperature drop, air to air}} = \frac{\frac{1}{1.65}}{1.27}$$

$$\text{Temperature drop, inside air to surface} = \frac{90}{1.27 \times 1.65} = 43$$

$$70 - 43 = 27 \text{ F, inside surface temperature of wall.}$$

14 ● How many inches of insulating material having a conductivity of 0.30 would be required, for the wall of Question 3, to raise the inside surface temperature to 60 F?

Temperature drop, air to inside surface = 10 F; temperature drop, inside surface to outside air = 80 F. Therefore, the heat resistance from inside wall surface to outside air must be eight times that from inside air to inside wall surface, or $8 \times \frac{1}{1.65} = 4.85$. The resistance for added material is, therefore,

$$4.85 - \left(\frac{6}{12} + \frac{1}{6}\right) = 4.19$$

$$4.19 \times 0.30 = 1.25 \text{ in. of insulation.}$$

Chapter 6

AIR LEAKAGE

Nature of Air Infiltration, Air Leakage Through Walls, Window Leakage, Wind Velocity to be Selected, Crack used for Computations, Multi-Story Buildings, Heat Equivalent of Air Entering by Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 7 and 8.)

THE NATURE OF AIR FILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

The wind causes a pressure to be exerted on one or two sides of a building. As a result, air comes into the building on the windward side through cracks or porous construction, and a similar quantity of air leaves on the leeward side through like openings. In general the resistance to air movement is similar on the windward to that on the leeward side. This causes a building up of pressure within the building and a lesser air leakage than that experienced in single wall tests as determined in the laboratory. It is assumed that actual building leakages owing to this building up of pressure will be 80 per cent of laboratory test values. While there are cases where this is not true, tests in actual buildings substantiate the factor for the general case. Tests on mechanically ventilated classrooms of average construction have shown that air infiltration acts quite independently of the planned air supply. Accordingly, the heating or cooling load owing to air infiltration from natural causes should be considered in addition to the ventilating load.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-storied buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

AIR LEAKAGE THROUGH WALLS

Table 1[1] gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

TABLE 1. INFILTRATION THROUGH WALLS

Expressed in cubic feet per square foot per hour[a]

TYPE OF WALL		WIND VELOCITY, MILES PER HOUR					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12.2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in A.S.H.V.E. research papers entitled Air Infiltration Through Various Types of Brick Wall Construction, and Air Infiltration Through Various Types of Wood Frame Construction. (See References on pages 128 and 129).

[b]Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

[1]Air Infiltration through Various Types of Brick Wall Construction, by Larson, Nelson and Braatz A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

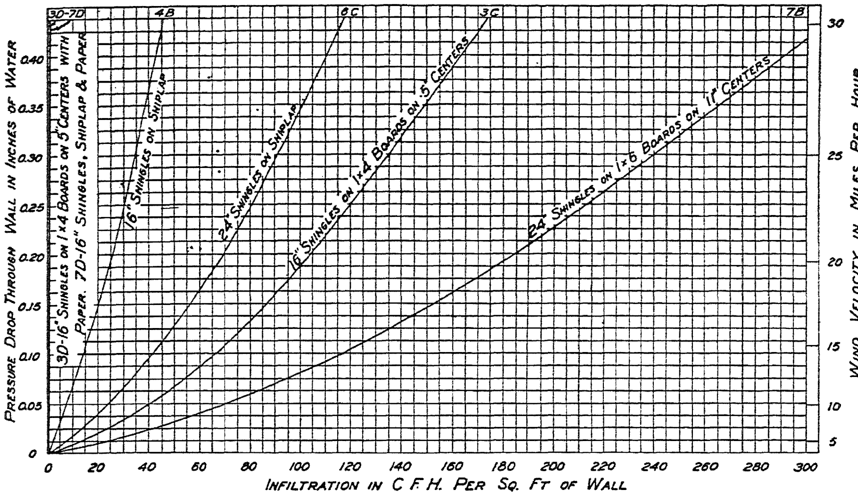

FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only, without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

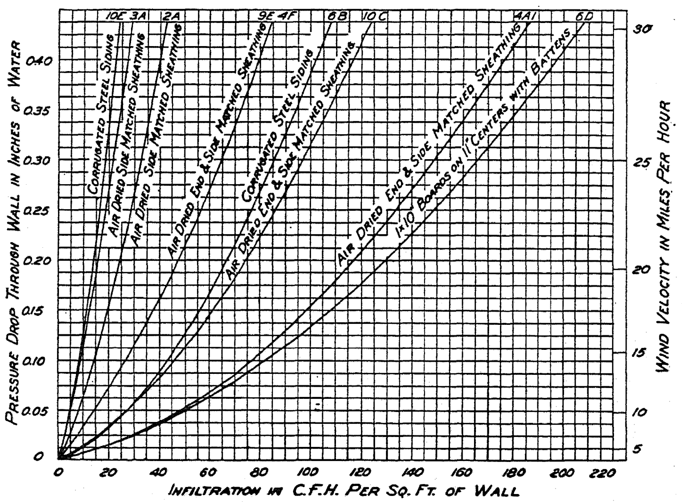

FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

The amount of infiltration that may be expected through simple walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration there indicated is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance as illustrated in Fig. 3. The length of the perimeter opening or crack for a double-hung window is equal to three times the width plus two times the height, or in other words, it is the outer sash perimeter length plus the meeting rail length. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For metal pivoted sash, the length of crack is the total perimeter of the movable or ventilating sections. Frame leakage on steel windows may be neglected when they are properly grouted with cement mortar into brick work or concrete. When they are not properly sealed, the linear feet of sash section in contact with steel work at mullions should be figured at 25 per cent of the values for industrial pivoted windows as given in Table 2.

Leakage values for storm sash are given in Figs. 4 and 5. When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

Doors vary greatly in fit because of their large size and tendency to warp. For a well fitted door, the leakage values for a poorly fitted double-hung wood window may be used. If poorly fitted, twice this figure should

TABLE 2. INFILTRATION THROUGH WINDOWS

Expressed in Cubic Feet per Foot of Crack per Hour[a]

TYPE OF WINDOW	REMARKS	WIND VELOCITY, MILES PER HOUR					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance[c]. Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance[e]. Includes wood frame leakage[d].	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted,[g] 1/16-in. crack	52	108	176	244	304	372
	Architectural projected,[h] 3/64-in. crack	20	52	88	116	152	208
	Residential casement,[i] 1/32-in. crack	14	32	52	76	100	128
	Heavy casement section, projected,[j] 1/32-in. crack	8	24	38	54	72	96
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

be used. If weatherstripped, the values may be reduced one-half. A single door which is frequently opened, such as might be found in a store, should have a value applied which is three times that for a well fitted door. This extra allowance is for opening and closing losses and is kept from being greater by the fact that doors are not used as much in the coldest and windiest weather.

CHOOSING WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Until this point is definitely established the practice of using average values will be followed. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 7.

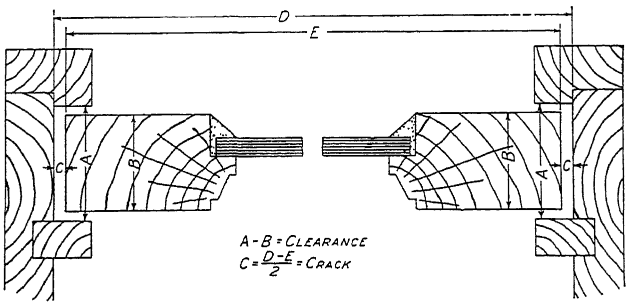

FIG. 3. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, as already explained, is to select an outside temperature based on the lowest on record and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing

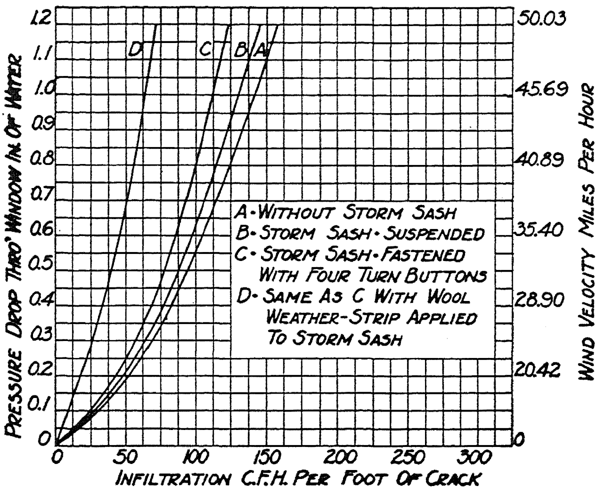

FIG. 4. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—5/64-IN. CRACK AND 1/32-IN. CLEARANCE

and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 4.)

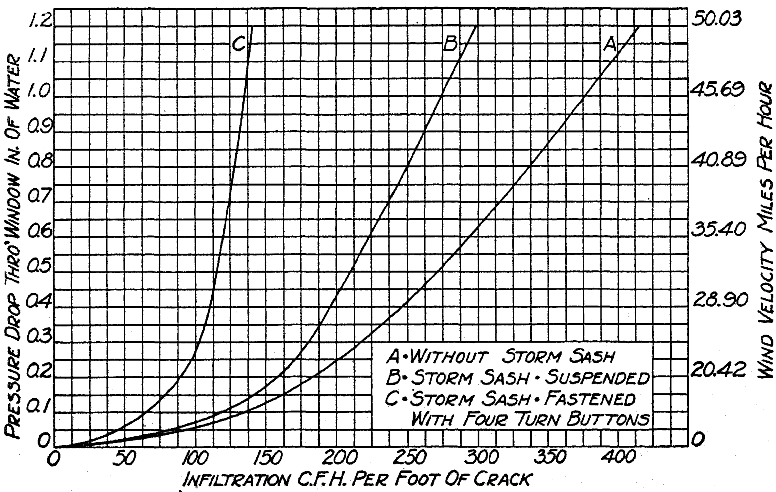

FIG. 5. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH—1/8-IN. CRACK AND 1/8-IN. CLEARANCE

CRACK USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 3. This method may be used to advantage as a check on the calculations made in the more exact manner.

Table 3. Air Changes Taking Place under Average Conditions Exclusive of Air Provided for Ventilation

Kind of Room or Building	Number of Air Changes Taking Place per Hour
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc	½ to 3

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels.[2] On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and temperature difference may seldom coöperate to the fullest extent. Making the rough assumption that the *neutral zone* is located at mid-

[2]Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, August, 1934, p. 349).

height of a building, and that the temperature difference is 70 F, the following formulae may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \qquad (1)$$

$$M_e = \sqrt{M^2 + 1.75\,b} \qquad (2)$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height.
b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist, although probably no greater wind velocities should be figured at such extremely high levels[3].

Sealing of Vertical Openings[4]

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead, the greater wind movement at the high altitudes makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

Heating Surface for Stair-Wells[4]

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the stratification of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual

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[3]Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw, and Carl Gutberlet (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, September, 1934).
[4]See Flue Action in Tall Buildings, by H. L. Alt (*Heating, Piping and Air Conditioning*, May, 1932).

way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR ENTERING BY INFILTRATION

The heat required to warm cold, outside air, which enters a room by infiltration, to the temperature of the room is given by the following equation:

$$H_i = 0.24\ Q\ d\ (t - t_o) \qquad (3)$$

where

H_i = Btu per hour required for heating air leaking into building from outside temperature t_o to inside temperature t.

Q = cubic feet of air entering per hour at inside temperature t.

d = density (pounds per cubic foot) of air at inside temperature t.

t = inside temperature at the proper level.

t_o = outside air temperature for which heating system is designed.

0.24 = specific heat of air.

It is sufficiently accurate to take d = 0.075 lb, in which case the equation reduces to

$$H_i = 0.018\ Q\ (t - t_o) \qquad (4)$$

While a heating reserve must be provided to warm inleaking air on the windward side of a building, this does not necessarily mean that the heating plant must be provided with a reserve capacity, since the inleaking air, warmed at once by adequate heating surface in exposed rooms, will move transversely and upwardly through the building, thus relieving other radiators of a part of their load. The actual loss of heat of a building caused by infiltration is not to be confused with the necessity for providing additional heating capacity for a given space. Infiltration is a disturbing factor in the heating of a building, and its maximum effect (maximum in the sense of an average of wind velocity peaks during the heating season above some reasonably chosen minimum) must be met by a properly distributed reserve of heating capacity, which reserve, however, is not in use at all places at the same time, nor in any one place at all times.

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Air Infiltration Through Steel Framed Windows, by D. O. Rusk, V. H. Cherry and L. Boelter (*Heating, Piping and Air Conditioning*, October, 1932).

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PROBLEMS IN PRACTICE

1 ● What are the causes of infiltration (or exfiltration) and how do they act on a building?

The wind and temperature differences create differences between internal and external pressures which cause air to flow through any openings in the walls.

2 ● Why is it essential to consider this in heating calculations?

The inflowing air displaces inside heated air and must be heated up to the internal temperature.

3 ● Where is it necessary to consider infiltration created by temperature difference?

In tall, single-story buildings and in multi-story buildings where the floors are not adequately isolated.

4 ● Why is the infiltration in a building less than that determined in laboratory tests?

In laboratory tests, the indicated wind velocity is measured by the difference in pressure on the two sides of a single wall, window, or object tested. In a building, an internal back pressure is built up between its walls to a point where outflow on the lee side is equal to inflow on the windward side and this back pressure reduces the actual inflow below that determined in the laboratory for a comparable wind.

5 ● Is heat loss by infiltration through walls of importance?

Only in the case of simple walls or poorly constructed compound walls.

6 ● What measurements are required to calculate the heat loss through double-hung wood windows?

Sash crack (equal to the sash perimeter plus the meeting rail) and frame crack (equal to the frame perimeter).

7 ● What is the basis for selecting the wind velocity and outside temperature to be used in making infiltration calculations?

Weather Bureau records. The wind velocity taken is the average during the three coldest months and the temperature used is the lowest on record for the given locality.

8 ● How does the temperature difference influence the heat loss in a tall building?

The chimney effect caused by the temperature difference operates to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels.

9 ● For a wind velocity of 15 mph and a building 180 ft high, calculate the effective wind velocity at the ground floor and at a height of 150 ft.

a. At the ground floor the effective wind velocity would be

$$M_e = \sqrt{15^2 + 1.75 \times 90} = 19.6 \text{ mph}$$

b. At a floor 150 ft above the ground

$$M_e = \sqrt{15^2 - 175 \times 60} = 11.0 \text{ mph}$$

10 ● A room contains three 2 ft-8 in. by 5 ft-6 in. plain double-hung wood windows with 1/16-in. crack and 3/64-in. clearance. Assume a wind velocity of 20 mph and a temperature difference of 75 F. Neglecting chimney effect, what is the maximum heat loss due to infiltration?

From Table 2, heat loss per foot of crack per degree temperature difference is 1.067 Btu per hour. Length of crack for the three windows is 57 ft. The maximum heat loss, due to infiltration, is equal to 1.067 × 57 × 75 or 4561 Btu per hour.

11 ● Find the infiltration through a wall with 16-in. shingles on 1 in. by 4 in. boards with 20 mph wind velocity. Give the pressure drop through the wall.

Referring to Curve 3C, Fig. 1, the value on the horizontal scale corresponding to 20 mph is approximately 102 cfh per square foot of wall.

The pressure drop through the wall is 0.193 in. of water (see left hand vertical scale).

12 ● What will be the infiltration through air-dried end and side-matched sheathing for 15 mph wind velocity?

Referring to Curve 10C, Fig. 2, the value on the horizontal scale corresponding to 15 mph is 50 cfh per square foot of wall.

13 ● From Table 2, find the infiltration (cubic feet per hour per foot of crack) for an average double-hung window, not weather stripped, with a 20 mph wind velocity.

59.3 cu ft per foot of crack per hour.

14 ● Using the value found in Question 11, what will be the heat requirement in a building with a total crack (all windows and doors) of 180 ft if the wind velocity is 15 mph, the outside temperature is 0 F, and the inside temperature is 70 F?

Using one half of the total crack, the volume of air is:

$$90 \times 59.3 = 5337 \text{ cu ft}$$

$$H = 0.018 \times 5337 \times (70 - 0) = 6724.6 \text{ Btu. (See Equation 4.)}$$

Chapter 7

HEATING LOAD

Factors Governing Heat Demand, Procedure, Temperatures, Wind Movement, Heat Sources Other Than Heating Plant, Example, Condensation

TO design any system of heating, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be capable of maintaining the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

1. Outside temperature. 2. Rain or snow. 3. Sunshine or cloudiness. 4. Wind velocity.	*Outside Conditions (The Weather)*
5. Heat transmission of exposed parts of building. 6. Infiltration of air through cracks, crevices and open doors and windows. 7. Heat capacity of materials. 8. Rate of absorption of solar radiation by exposed materials.	*Building Construction*
9. Inside temperatures. 10. Stratification of air. 11. Type of heating system. 12. Ventilation requirements. 13. Period and nature of occupancy. 14. Temperature regulation.	*Inside Conditions*

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time, while desirable, is hard of attainment.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1.)

2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself. (See Table 2.)

3. Select or compute the heat transmission coefficients for outside walls and glass; also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 5.)

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See Items 1 and 2.)

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 6.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions re-

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

Type of Building	Deg Fahr	Type of Building	Deg Fahr
SCHOOLS		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

[a]The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. See Chapter 2.

quire, but inasmuch as the heating system proportioned for taking care of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or have more time allowed for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 2. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.[1]

According to Fig. 2, Chapter 2, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures given in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat-loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of pitched roofs, unheated attics and top-floor ceilings Chapter 5.)

[1]See Chapter 2, p. 43.

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight-line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information.

Where ceiling heights are under 20 ft, it is common engineering practice to consider that the Fahrenheit temperature increases 2 per cent for each foot of height above the breathing line. This rule, sufficiently accurate for most cases, will give the probable air temperature at any given level for a room heated by direct radiation. Thus, the probable temperature in a room at a point three feet above the breathing line, if the breathing line temperature is 70 F, will be

$$(1.00 + 3 \times .02)\ 70 = 74.2 \text{ F.}$$

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters, air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling by a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing-line temperature may be used, provided the breathing-line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat-loss computations.

The outside temperature to be assumed in the design of any heating system is ordinarily not more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In

the case of massive and well insulated buildings in localities where the minimum does not prevail for more than a few hours, or where the lowest recorded temperature is extremely unusual, more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau at the places listed.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND MOVEMENT

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.

2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity which records show to have occurred and to select the worst combination; but designers generally do not feel that such a degree of refinement is justified. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 5, by using a surface coefficient f_o for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 5 are based.

In a similar manner, the heat allowance for infiltration through cracks and walls (Tables 1 and 2, Chapter 6) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 6).

In the past many designers have used empirical *exposure factors* which

Table 2. Climatic Conditions Compiled from Weather Bureau Records

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
State	City	Average Temp., Oct. 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Mobile	57.7	−1	8.3	N
	Birmingham	53.9	−10	8.6	N
Ariz.	Phoenix	59.5	12	3.9	E
	Flagstaff	34.9	−25	6.7	SW
Ark.	Fort Smith	49.5	−15	8.0	E
	Little Rock	51.6	−12	9.9	NW
Calif.	San Francisco	54.3	27	7.5	N
	Los Angeles	58.6	28	6.1	NE
Colo.	Denver	39.3	−29	7.4	S
	Grand Junction	39.2	−21	5.6	SE
Conn.	New Haven	38.0	−15	9.3	N
D. C.	Washington	43.2	−15	7.3	NW
Fla.	Jacksonville	61.9	10	8.2	NE
Ga.	Atlanta	51.4	−8	11.8	NW
	Savannah	58.4	8	8.3	NW
Idaho	Lewiston	42.5	−23	4.7	E
	Pocatello	36.4	−22	9.3	SE
Ill.	Chicago	36.4	−23	17.0	SW
	Springfield	39.9	−24	10.2	NW
Ind.	Indianapolis	40.2	−25	11.8	S
	Evansville	44.1	−16	8.4	S
Iowa	Dubuque	33.9	−32	6.1	NW
	Sioux City	32.1	−35	12.2	NW
Kans.	Concordia	38.9	−25	7.3	N
	Dodge City	40.2	−26	10.4	NW
Ky.	Louisville	45.2	−20	9.3	SW
La.	New Orleans	61.5	7	9.6	N
	Shreveport	56.2	−5	7.7	SE
Me.	Eastport	31.1	−23	13.8	W
	Portland	33.6	−21	10.1	NW
Md.	Baltimore	43.6	−7	7.2	NW
Mass.	Boston	37.6	−18	11.7	W
Mich.	Alpena	29.1	−28	11.3	W
	Detroit	35.4	−24	13.1	SW
	Marquette	27.6	−27	11.4	NW
Minn.	Duluth	25.1	−41	11.1	SW
	Minneapolis	29.6	−33	11.5	NW
Miss.	Vicksburg	56.0	−1	7.6	SE
Mo.	St. Joseph	40.3	−24	9.1	NW
	St. Louis	43.3	−22	11.8	NW
	Springfield	43.0	−29	11.3	SE
Mont.	Billings	34.7	−49	....	W
	Havre	27.7	−57	8.7	SW
Nebr.	Lincoln	37.0	−29	10.9	N
	North Platte	34.6	−35	9.0	W
Nev.	Tonopah	39.6	−10	9.9	SE
	Winnemucca	37.9	−28	9.5	NE
N. H.	Concord	33.4	−35	6.0	NW
N. J.	Atlantic City	41.6	−9	10.6	NW
N. Y.	Albany	35.1	−24	7.9	S
	Buffalo	34.7	−20	17.7	W
	New York	40.7	−14	17.1	NW
N. M.	Santa Fe	38.0	−13	7.3	NE

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS—(Continued)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State or Province	City	Average Temp., Oct. 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
N. C.	Raleigh	49.7	−2	7.3	SW
	Wilmington	53.1	5	8.9	SW
N. Dak.	Bismarck	24.5	−45	....	NW
	Devils Lake	18.9	−44	11.4	W
Ohio	Cleveland	36.9	−17	14.5	SW
	Columbus	39.9	−20	9.3	SW
Okla.	Oklahoma City	48.0	−17	12.0	N
Oreg.	Baker	34.1	−24	6.0	SE
	Portland	45.9	−2	6.5	S
Pa.	Philadelphia	41.9	−6	11.0	NW
	Pittsburgh	40.8	−20	13.7	NW
R. I.	Providence	37.6	−17	14.6	NW
S. C.	Charleston	56.9	7	11.0	N
	Columbia	53.7	−2	8.0	NE
S. Dak.	Huron	28.1	−43	11.5	NW
	Rapid City	32.3	−34	7.5	W
Tenn.	Knoxville	47.0	−16	6.5	SW
	Memphis	50.9	−9	9.6	NW
Texas	El Paso	53.0	−2	10.5	NW
	Fort Worth	54.7	−8	11.0	NW
	San Antonio	60.7	4	8.2	N
Utah	Modena	38.1	−24	8.9	W
	Salt Lake City	40.0	−20	4.9	SE
Vt.	Burlington	29.3	−28	12.9	S
Va.	Norfolk	49.1	2	9.0	N
	Lynchburg	45.2	−7	5.2	NW
	Richmond	47.4	−3	7.4	S
Wash.	Seattle	45.3	3	9.1	SE
	Spokane	37.5	−30	5.2	SW
W. Va.	Elkins	38.8	−28	4.8	W
	Parkersburg	41.9	−27	6.6	S
Wis.	Green Bay	28.6	−36	12.8	SW
	La Crosse	31.2	−43	5.6	NW
	Milwaukee	33.0	−25	11.7	W
Wyo.	Sheridan	31.0	−45	5.3	NW
	Lander	28.9	−40	3.0	NE
Alta.	Edmonton	23.3	−57	4.5	W
B. C.	Victoria	43.8	−2	8.9	N
	Vancouver	41.7	2	4.2	E
Man.	Winnipeg	17.2	−46	12.4	SW
N. B.	Fredericton	27.1	−35	8.7	NW
N. S.	Yarmouth	35.5	−12	13.0	NW
Ont.	London	32.5	−26	....	....
	Ottawa	26.9	−33	7.5	W
	Pt. Arthur	21.6	−51	....	....
	Toronto	32.0	−28	13.5	SW
P. E. I.	Charlottetown	30.1	−27	8.7	NW
Que.	Montreal	27.4	−27	15.4	SW
	Quebec	24.4	−34	15.0	SW
Sask.	Prince Albert	14.7	−70	3.2	SW
Yukon	Dawson	1.6	−68	....	....

were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities. Recent investigations indicate, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

Pending the time when the lack of actual test data is remedied, it is recommended that no differentiation be made among the various sides of a building in calculating the heat losses. It should be remembered that the values for U in the tables in Chapter 5 are based on a wind velocity of 15 mph.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method[2] for calculating the square feet of equivalent direct radiation required in a building. This method makes use of exposure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and the velocity of them.

HEAT FROM SOURCES OTHER THAN HEATING PLANT

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \dfrac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2{,}546$, and in the second case Btu per hour = bhp $\times$ 2,546, in which 2,546 is the Btu equivalent of 1 hp-hour. In high-powered mills this is the chief

[2]See Standards of *Heating, Piping and Air Conditioning Contractors National Association.*

source of heating and it is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.415. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas gives off about 535 Btu per hour; and one cubic foot of natural gas gives off about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 2.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 2.)

CONDENSATION ON BUILDING SURFACES[3]

Condensation on the interior surfaces of buildings is often a serious problem. Water dripping from a ceiling may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power and lighting systems, necessitating shut-downs and incurring costly repairs. It also causes rotting of wood roof structures, corrosion of metal roofs, and spalling and disintegration of gypsum and other types of roof decks not properly protected.

Condensation is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature, and can be remedied in two ways, (1) by increasing the temperature of such surfaces above the dew-point temperature, or (2) by lowering the humidity.

Dehumidification, of course, is not advisable where a high relative humidity is necessary for manufacturing processes. Hence, the only alternative is to increase the surface temperature by decreasing the inside surface resistance. This can be accomplished by increasing the velocity of air passing over the surface, or by increasing the over-all resistance of the wall or roof by installing a sufficient thickness of insulation.

The latter method is generally used, and the thickness of insulation is determined by ascertaining the amount of resistance to be added to increase the temperature of the interior surface above the dew-point temperature for the maximum conditions involved. This in turn is based on the fundamental principle that the drop in temperature is proportional to the resistance. See Question 1 at the end of this chapter.

[3]See Preventing Condensation on Interior Building Surfaces, by Paul D. Close (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

EXAMPLE OF HEAT LOSS COMPUTATIONS

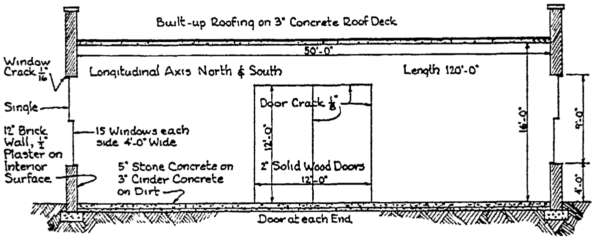

FIG. 1. ELEVATION OF FACTORY BUILDING

1. LOCATION..Philadelphia, Pa.
2. LOWEST OUTSIDE TEMPERATURE. (Table 2).. − 6 F
3. BASE TEMPERATURE: *In this example* a design temperature 10 F above lowest on record instead of 15 F is used. Hence the base temperature =

$$(-\ 6 + 10) = +\ 4\ \text{F}.$$

4. DIRECTION OF PREVAILING WIND (during Dec., Jan., Feb.)....................Northwest
5. BREATHING-LINE TEMPERATURE (5 ft from floor)..60 F
6. INSIDE AIR TEMPERATURE AT ROOF:

   The air temperature just below *roof* is higher than at the breathing line. Height of roof is 16 ft, or it is 16 − 5 = 11 ft above breathing line. Allowing 2 per cent per foot above 5 ft, or 2 × 11 = 22 per cent, makes the temperature of the air under the roof = 1.22 × 60 = 73.2 F.

7. INSIDE TEMPERATURE AT WALLS:

   The air temperature at the mean height of the *walls* is greater than at the breathing line. The mean height of the walls is 8 ft and allowing 2 per cent per foot above 5 ft, the average mean temperature of the walls is 1.06 × 60 = 63.6 F. By similar assumptions and calculations, the mean temperature of the *glass* will be found to be 64.2 F and that of the *doors* 61.2 F.

8. AVERAGE WIND VELOCITY (Table 2)..11.0 mph
9. OVER-ALL DIMENSIONS (See Fig. 1)..120 x 50 x 16 ft
10. CONSTRUCTION:

    Walls—12-in. brick, with ½-in. plaster applied directly to inside surface.
    Roof—3-in. stone concrete and built-up roofing.
    Floor—5-in. stone concrete on 3-in. cinder concrete on dirt.
    Doors—One 12 ft x 12 ft wood door (2 in. thick) at each end.
    Windows—Fifteen, 9 ft x 4 ft single glass double-hung windows on each side.

11. TRANSMISSION COEFFICIENTS:

    Walls—(Table 3, Chapter 5, Wall 2B)........................U = 0.34
    Roof—(Table 11, Chapter 5, Roofs 2A and 3A)........................U = 0.77
    Floor—(Table 10, Chapter 5, Floors 5A and 6A)........................U = 0.63
    Doors—(Table 13B, Chapter 5)........................U = 0.46
    Windows—(Table 13A, Chapter 5)........................U = 1.13

12. INFILTRATION COEFFICIENTS:

Windows—Average windows, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance. The leakage per foot of crack for an 11-mile wind velocity is 25.0 cfh. (Determined by interpolation of Table 2, Chapter 6.) The heat equivalent per hour per degree per foot of crack is taken from Chapter 6.

$25.0 \times 0.018 = 0.45$ Btu per deg Fahr per foot of crack.

Doors—Assume infiltration loss through door crack twice that of windows or $2 \times 0.45 = 0.90$ Btu per deg Fahr per foot of crack.

Walls—As shown by Table 1, Chapter 6, a plastered wall allows so little infiltration that in this problem it may be neglected.

13. CALCULATIONS: See calculation sheet, Table 3.

TABLE 3. CALCULATION SHEET SHOWING METHOD OF ESTIMATING HEAT LOSSES OF BUILDING SHOWN IN FIG. 1

PART OF BUILDING	WIDTH IN FEET	HEIGHT IN FEET	NET SURFACE AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIFF.	TOTAL BTU
North Wall:						
Brick, ½-in. plaster	50	16	656	0.34	59.6	13,293
Doors (2-in. wood)	12	12	144	0.46	57.2	3,789
⅛ in. Crack	1 pair doors		60	0.90	57.2	1,544[a]
West Wall:						
Brick, ½-in plaster	120	16	1380	0.34	59.6	27,964
Glass (Single)	15 x 4	9	540	1.13	60.2	36,734
1/16 in. Crack	Double Hung Windows (15)		450	0.45	60.2	6,095[a]
South Wall	Same as North Wall					18,626
East Wall	Same as West Wall					70,793
Roof, 3-in. concrete and slag-surfaced built-up roofing	50	120	6000	0.77	69.2	319,704
Floor, 5-in. stone concrete on 3-in. cinder concrete	50	120	6000	0.63	5[b]	18,900
GRAND TOTAL of heat required for building in Btu per hour						517,442

[a]This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack will be used in computing infiltration for each side and each end of building.

[b]A 5 F temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground.

PROBLEMS IN PRACTICE

1 ● The dry-bulb temperature and the relative humidity at the ceiling of a mixing room in a bakery are 80 F and 60 per cent, respectively. The roof is a 4-in. concrete deck covered with built-up roofing. If the lowest outside temperature to be expected is −10 F, what thickness of rigid fiber insulation will be required to prevent condensation?

From Table 11, Chapter 5, U for the uninsulated roof = 0.72. From Table 2, Chapter 5, k for rigid fiber insulation = 0.33. From the psychrometric chart, Chapter 1, the dew point of air at 80 F and 60 per cent relative humidity is 65 F. The ceiling temperature, therefore, must not drop below 65 F if condensation is to be prevented.

When equilibrium is established, the amount of heat flowing through any component part of a construction is the same for each square foot of area.

Therefore,

$$U\,[80 - (-10)] = 1.65\,(80 - 65)$$

where

U is the transmittance of the insulated roof.

Solving the equation, $U = 0.275$.

The resistance of the insulated roof $= \dfrac{1}{0.275} = 3.64.$

The resistance of the uninsulated roof $= \dfrac{1}{0.72} = 1.39.$

The resistance of the insulation $= 3.64 - 1.39 = 2.25.$

Resistance per inch of insulation $= \dfrac{1}{0.33} = 3.0.$

Since a resistance of 2.25 is required, and 1 in. of insulation has a resistance of 3, one inch will be sufficient to prevent condensation.

The same result might have been obtained by selecting an insulated 4-in. concrete slab having a U of less than 0.275 from Table 11, Chapter 5. This 4-in. concrete slab with 1-in. rigid insulation has a U of 0.23 which is safe.

2 ● What inside dry-bulb temperatures are usually assumed for: (a) homes, (b) schools, (c) public buildings?

Referring to Table 1:

a. 70 to 72 F.

b. Temperature varies from 55 to 75 F, depending on the room. Classrooms, for instance, are usually specified as 70 to 72 F.

c. 68 to 72 F.

3 ● How is the outside temperature selected for use in computing heat losses?

The outside temperature used in computing heat losses is generally taken from 10 to 15 F higher than the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In some cases where the lowest recorded temperature is extremely unusual, the design temperature is taken even higher than 15 F above the lowest recorded temperature.

4 ● What are the effects of wind movement on the heating load?

a. Wind movement increases the heat transmission of walls, glass, and roof; it affects poor walls to a much greater extent than good walls.

b. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves if such materials are at all porous.

5 ● Calculate the heat given off by eighteen 200-watt lamps.

$200 \times 18 \times 3.415 = 12{,}294$ Btu per hour.

6 ● A two-story, six room, frame house, 28-ft by 30-ft foundation, has the following proportions:

Area of outside walls, 1992 sq ft.
Area of glass, 333 sq ft.
Area of outside floors, 54 sq ft.
Cracks around windows, 440 ft.
Cracks around doors, 54 ft.
Area of second floor ceiling, 783 sq ft.
Volume, first and second floors, 13,010 cu ft.
Ceilings, 9 ft high.

The minimum temperature for the heating season is −34 F, and the required inside temperature at the 30-in. level is 70 F. The average number of degree days for a heating season is 7851, and the average wind velocity is 10 mph, northwest.

The walls are constructed of 2-in. by 4-in. studs with wood sheathing, building paper, and wood siding on the outside, and wood lath and plaster on the inside. Windows are single glass, double-hung, wood, without weatherstrips. The second floor ceiling is metal lath and plaster, without an attic floor. The roof is of wood shingles on wood strips with rafters exposed. The area of the roof is 20 per cent greater than the area of the ceiling. Select values for the following: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for roof; (e) U for ceiling and roof combined; (f) air leakage, cubic feet per hour per foot of window crack; (g) air leakage, cubic feet per hour per foot of door crack.

a. 0.25 (Table 5, Chapter 5).
b. 1.13 (Table 13, Chapter 5).
c. 0.69 (Table 8, Chapter 5).
d. 0.48 (Table 12, Chapter 5).
e. 0.236 (Equation 6, Chapter 5).
f. 21.4 (Table 2, Chapter 6).
g. 42.8, which is double the window leakage.

7 ● Using the data of Question 6, calculate the maximum Btu loss per hour for the various constructions, and show the percentage of the total heat which is lost through each construction described.

Assume 2 per cent rise in temperature for each foot in height. The average temperature will be 72.8 F for walls, doors, and windows, and 79.1 F for the second floor ceiling.

a. Outside walls	46,200 Btu loss	37.2 per cent of total
b. Glass	34,950 Btu loss	28.1 per cent of total
c. Doors	5,670 Btu loss	4.6 per cent of total
d. Second floor ceiling	17,840 Btu loss	14.3 per cent of total
e. Air leakage, windows	15,750 Btu loss	12.7 per cent of total
f. Air leakage, doors	3,865 Btu loss	3.1 per cent of total
Total	124,275 Btu loss	100.0 per cent of total

8 ● For the house in Question 6, place 1-in. insulation in the outside walls and second floor ceiling; k for insulation = 0.34. Use weatherstrip on doors and windows, and double glass on the windows; C_a = 0.55. Calculate or select the following values: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for combination of ceiling and roof; (e) Air leakage, cubic feet per hour per foot of door crack; (f) air leakage, cubic feet per hour per foot of window crack.

a. 0.144.
b. 0.55.
c. 0.23.
d. 0.13.
e. 15.5.
f. 31.0.

9 ● Calculate the maximum Btu loss per hour and show the percentage loss by each channel for the house as insulated in Question 8.

a. Outside walls	26,650 Btu loss	36.2 per cent of total
b. Glass	17,000 Btu loss	23.1 per cent of total
c. Doors	5,670 Btu loss	7.7 per cent of total
d. Ceiling	10,070 Btu loss	13.7 per cent of total
e. Air leakage, windows	11,400 Btu loss	15.5 per cent of total
f. Air leakage, doors	2,795 Btu loss	3.8 per cent of total
Total	73,585 Btu loss	100.0 per cent of total

10 ● From the results of Questions 7 and 9, calculate the Btu saved and the percentage saved by each change in construction.

	Insulated	Uninsulated	Btu Saved	Per Cent Saved
a. Outside walls	46,200	26,650	19,550	42.3
b. Glass	34,950	17,000	17,950	51.4
c. Doors	5,670	5,670	0	0
d. Ceiling	3,865	2,795	1,070	27.7
e. Air leakage, windows	17,840	10,070	7,700	43.1
f. Air leakage, doors	15,750	11,400	4,350	27.6

11 ● From the results of Questions 7 and 9, calculate the heat loads per heating season in Btu and note the savings by better construction.

The 7851 degree days for the heating season multiplied by 24 hours, times the Btu loss per hour for 1 F drop in temperature gives the Btu load per heating season.

Saving = 250,800,000 − 148,000,000 = 102,800,000 Btu.

Chapter 8

COOLING LOAD

Conditions to be Maintained, Cooling Load, Transmission for Surfaces not Exposed to the Sun, Outside Temperatures, Solar Radiation, Time Lag, Transmission of Solar Radiation Through Glass, Heat and Moisture Leakage, Heat and Moisture Sources

THE method of calculating the cooling load is similar to that used in calculating the heating load. The direction of the flow of heat is reversed, however, and in most cases additional factors must be considered, such as solar radiation and the heat from occupants, lights, motors, and other sources. The character of the load depends on the type of building to be cooled as, for example, in auditoriums and other places of assemblage where the maximum load usually is that due to the heat and moisture given off by the occupants, or in office buildings and residences where solar radiation and the transmission and infiltration of heat through the building shell are most important.

While cooling is generally identified with the summer season, it is often necessary to cool in winter as well as in summer. In a crowded place of assemblage the heat given off by the occupants, together with that given off by the lighting and power equipment, may be more than the normal heat loss through the structure even in winter under cold climatic conditions.

Much of the basic information for the design of comfort conditioning installations has resulted from research conducted at the A.S.H.V.E. Research Laboratory and at institutions with which coöperative research investigations have been carried on. These data include the effective temperature index, and heat and moisture loss data given in Chapter 2.

COMFORT CONDITIONS

The conditions to be maintained in an enclosure are variable and depend on many factors, especially the season of the year and (during the summer) the outside dry-bulb temperature and the duration of the period of occupancy. Information concerning the proper effective temperatures to be maintained for various seasons is given in Chapter 2, where are also tabulated the most desirable indoor air conditions to be maintained in summer for exposures less than three hours. (See Table 2, Chapter 2.)

In installations for restaurants and theaters the requirements are different from those in offices, since there must be a considerable volume of air circulated in order to provide ventilation and cooling.

TABLE 1. AVERAGE MAXIMUM DESIGN DRY-BULB TEMPERATURES, DESIGN WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

State	City	Average Maximum Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Ala.	Birmingham	93	77	5.2	S
	Mobile	94	78	8.6	SW
Ariz.	Phoenix	110	77	6.0	W
Ark.	Little Rock	95	77	7.0	NE
Calif.	Los Angeles	88	70	6.0	SW
	San Francisco	85	68	11.0	SW
Colo.	Denver	90	64	6.8	S
Conn.	New Haven	88	74	7.3	S
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	94	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	91	75	7.3	NW
	Savannah	95	79	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	91	75	8.2	S
Ind.	Indianapolis	90	73	9.0	SW
Iowa	Des Moines	92	74	6.6	SW
Ky.	Louisville	94	75	8.0	SW
La.	New Orleans	94	79	7.0	SW
Maine	Portland	85	71	7.3	S
Md.	Baltimore	93	76	6.9	SW
Mass.	Boston	88	73	9.2	SW
Mich.	Detroit	93	73	10.3	SW
Minn.	Minneapolis	84	72	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	92	75	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	87	63	7.3	SW
Nebr.	Lincoln	93	74	9.3	S
Nev.	Reno	93	64	7.4	W
N. J.	Trenton	95	76	10.0	SW
N. Y.	Albany	90	74	7.1	S
	Buffalo	83	72	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	87	63	6.5	SE
N. C.	Asheville	87	72	5.6	SE
	Wilmington	93	79	7.8	SW
N. Dak.	Bismarck	88	69	8.8	NW
Ohio	Cleveland	95	73	9.9	S
	Cincinnati	95	78	6.6	SW
Okla.	Oklahoma City	96	76	10.1	S
Oreg.	Portland	83	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	91	73	9.0	NW
R. I.	Providence	85	73	10.0	NW
S. C.	Charleston	94	80	9.9	SW
	Greenville	93	76	6.8	NE
Tenn.	Chattanooga	94	76	6.5	SW
	Memphis	93	77	7.5	SW

TABLE 1. AVERAGE MAXIMUM DESIGN DRY-BULB TEMPERATURES, DESIGN WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Continued)

STATE	CITY	AVERAGE MAXIMUM DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Texas	Dallas	99	76	9.4	S
	Galveston	93	79	9.7	S
	San Antonio	100	78	7.4	SE
	Houston	93	79	7.7	S
	El Paso	98	69	6.9	E
Utah	Salt Lake City	95	67	8.2	SE
Vt.	Burlington	85	71	8.9	S
Va.	Norfolk	91	76	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	83	61	7.9	S
	Spokane	89	63	6.5	SW
W. Va.	Parkersburg	90	74	5.3	SE
Wis.	Madison	89	73	8.1	SW
	Milwaukee	93	74	10.4	S
Wyo.	Cheyenne	85	62	9.2	S

COOLING LOAD

The cooling load may be divided into the following parts:

1. Transmission of heat through walls, roof, and glass with allowances for sun-exposed surfaces and heat capacity.
2. Transmission of solar radiation through glass and absorption by interior furnishings.
3. Heat and moisture from infiltration and from outside air introduced.
4. Heat and moisture from occupants and heat from lights, machinery and other sources.

Transmission for Surfaces Not Exposed to the Sun

The transmission load *for surfaces not exposed to the sun* is calculated in a manner similar to that described in Chapter 7, by means of the following formula:

$$H_t = A\,U\,(t_o - t) \qquad (1)$$

where

H_t = heat transmitted through the material of the wall, glass, roof, or floor, Btu per hour.
A = net inside area of wall, glass, roof, or floor, square feet.
t = inside temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
U = coefficient of transmission of wall, floor, roof, or glass, Btu per hour per square foot per degree Fahrenheit difference in temperature. (Tables 3 to 13, Chapter 5.)

Outside Temperatures

Summer dry-bulb and wet-bulb temperatures for various cities are given in Table 1. It will be noted that the temperatures are not the maximums but the design temperatures which should be used in air-conditioning calculations. The maximum outside wet-bulb temperatures

as given in Weather Bureau reports usually occur only from 1 per cent to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system covering this range. The temperatures shown in Table 1 have been chosen after extensive study of the Weather Bureau records and are temperatures that are not exceeded more than 5 to 8 per cent of the time during June, July, August, and September for an average year.

Solar Radiation

Fig. 1 shows the total amount of solar energy in Btu per square foot per hour received during the day by a surface normal to the rays of the sun, by a horizontal surface, and by east, west, and south walls. The curves are drawn from A.S.H.V.E. Laboratory data obtained by pyrheliometer, are based on sun time, and are for a perfectly clear day on August 1 at a north latitude of 40 deg. Data from these curves may be used with little error for most United States latitudes and for all of the hotter months of the year.

The absorption of solar radiation by a surface depends upon the character of the surface and the angle of the surface with respect to the direction of the radiation. The heat absorption by a black oilcloth surface perpendicular to the sun's rays was found to be as high as 273 Btu per square foot per hour, based on tests conducted by the A.S.H.V.E. Research Laboratory in Pittsburgh[1]. Lamp black, red brick dust, and aluminum bronze painted surfaces perpendicular to the sun's rays showed, respectively, 94.0, 63.4, and 28.2 per cent as high a rate of absorption as the black oilcloth.

TABLE 2. ALLOWANCE FOR SOLAR RADIATION ON ROOFS AND WALLS

APPROXIMATE NUMBER OF DEGREES TO ADD TO DRY-BULB TEMPERATURE FOR DIFFERENT TYPES OF SURFACES

TYPE OF SURFACE	BLACK	RED BRICK OR TILE	ALUMINUM PAINT
Roof, horizontal	45	30	15
East or west wall	30	20	10
South wall	15	10	5

Solar radiation is an important factor in the mechanism of heat flow into buildings. Research conducted at the A.S.H.V.E. Research Laboratory[2] has shown that a large error may be introduced into the calculations by failure to consider the periodical character of heat flow resulting from the diurnal movement of the sun and the heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall into a building on a hot, sunny day.

[1]Absorption of Solar Radiation in Relation to the Temperature, Color, Angle, and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[2]For further information on this subject see following A.S.H.V.E. research papers: Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928); Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930); Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Unfortunately, the calculations for the transmission of heat from solar radiation through building walls are too complicated to be of much practical value to the heating and ventilating engineer. Approximate results may be obtained by adding the number of degrees given in Table 2 to the outside design dry-bulb temperature in calculating the heat transmission through a wall or roof which may be exposed to the sun for an appreciable length of time. Table 2 was obtained from a study of the data in A.S.H.V.E. research papers on solar radiation[1, 3]. Black and aluminum painted surfaces represent the extremes which are likely to occur. For other types of surfaces, values intermediate between those given in the table can be used.

Time Lag

The calculation of heat transmitted through walls and roofs does not take into consideration the heat capacity of the structure and the consequent time lag in the transmission of heat. In the thick walls used in modern office buildings the time lag may amount to 10 hours or more[4]. Thus in many cases the wall transmission cannot be added directly to the cooling load from other sources because the peak of the wall transmission load may not coincide with the peak of the total cooling load and may even occur after the cooling system has been shut down for the day. The data in Table 3 were taken from A.S.H.V.E. research papers[3, 4] and while they result principally from a study of experimental slabs, they give an idea of the time lag to be expected in various structures.

TABLE 3. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

TYPE AND THICKNESS OF WALL OR ROOF	TIME LAG, HOURS
2-in. pine	1½
6-in. concrete	3
4-in. gypsum	2½
3-in. concrete and 1-in. cork	2
2-in. iron and cork (equivalent to ¾-in. concrete and 2.15-in. cork)	2½
4-in. iron and cork (equivalent to 5½-in. concrete and 1.94-in. cork)	7¼
8-in. iron and cork (equivalent to 16-in. concrete and 1.53-in. cork)	19
22-in. brick and tile wall	10

In intermittently cooled buildings an excess cooling capacity must be provided to care for the additional load imposed by the necessity to cool down the furnishings and the material of the interior construction to the point of maintained temperatures.

Transmission of Solar Radiation Through Glass

In considering the transmission through glass several factors must be considered. As the sun's rays impinge against a pane of glass, most of the radiation passes through to the other side, a small amount is reflected, and the balance is absorbed by the glass. The amount absorbed depends upon

[3]Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

[4]Field Studies of Office Building Cooling (A.S.H.V.E. Research Paper), by J. H. Walker, S. S. Sanford, and E. P. Wells (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

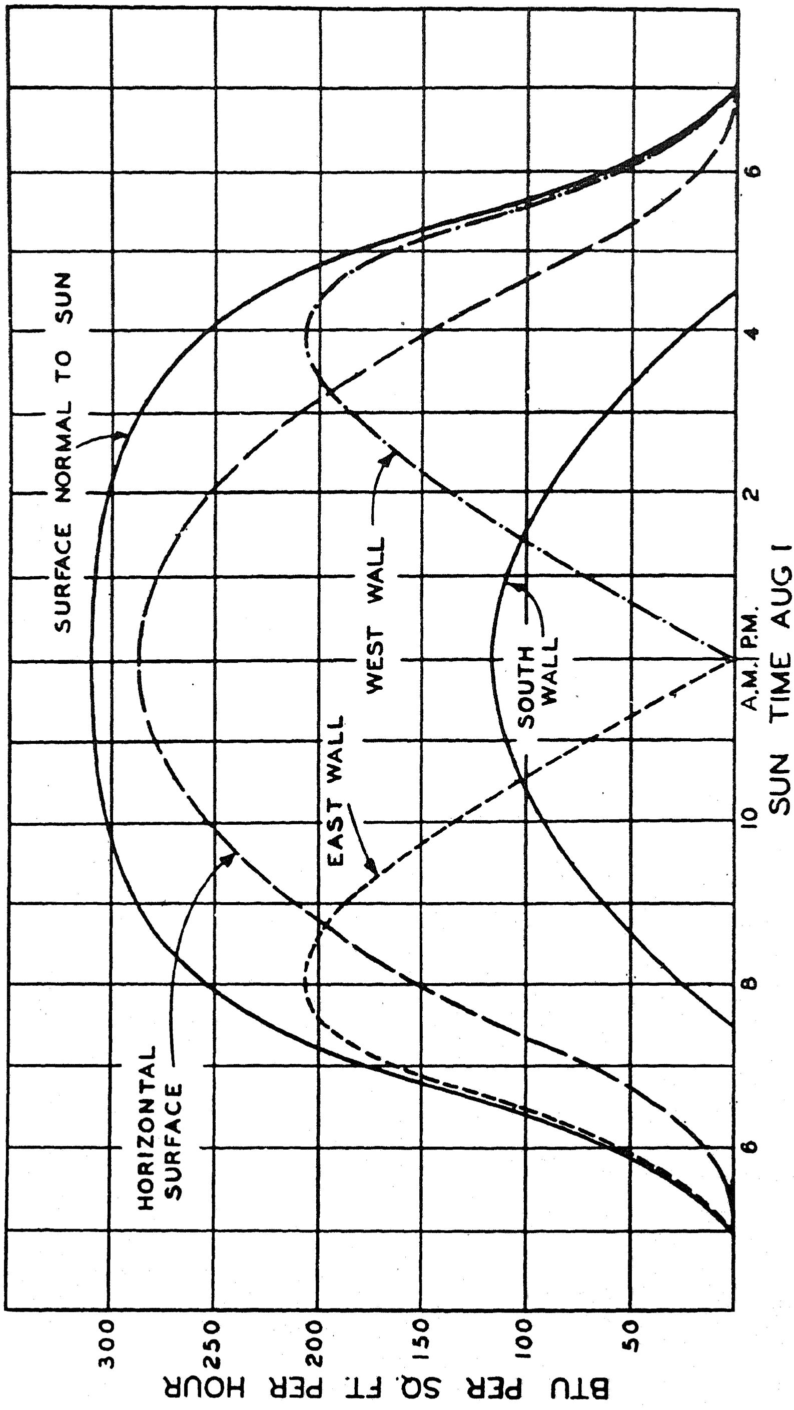

FIG. 1. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

the character and thickness of the glass and the angle between the rays of sunlight and the glass. The temperature of the glass is raised by the absorbed heat and this heat is then delivered to the air on the two sides of the glass in proportion to the difference between glass and air temperatures.

The A.S.H.V.E. tests indicated that a single pane of double strength glass 0.127 in. thick absorbs approximately 11 per cent of the solar radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle. Other experiments[4] indicate a glass absorption of 16.7 per cent for one pane of glass and 37.5 per cent for two ¼-in. panes separated by a 1¾-in. air space.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from the curves in Fig. 1. For surfaces other than those given, the solar radiation incident to the glass must be calculated. Hendrickson and Walker[5] have shown how this may be done if the wall faces some direction other than east, west, or south. They have also shown how to calculate the net glass area on which the solar radiation impinges when the glass is partly shaded by the frame or wall. The values from Fig. 1 must be used only for the net glass area on which the sun shines. Recent tests at the A.S.H.V.E. Research Laboratory[6] have determined the percentage of heat from solar radiation actually delivered to a room with bare windows and with various types of outdoor and indoor shading. The data in Table 4 are taken from these tests.

TABLE 4. SOLAR RADIATION TRANSMITTED THROUGH BARE AND SHADED WINDOWS

	PER CENT DELIVERED TO ROOM
Bare window glass	97
Canvas awning	28
Inside shade, fully drawn	45
Inside shade, one-half drawn	68
Inside Venetian blind, fully covering window	58
Outside Venetian blind, fully covering window	22

The percentage figures in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar radiation plus glass transmission based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass is small as the glass temperature is raised by the solar radiation absorbed. Therefore, in calculating the total heat gain through windows on the sunny sides of buildings, it is sufficiently accurate to figure the total cooling load due to the window, as the solar radiation times the proper factor from Table 4, and to neglect the heat

[5]Summer Cooling for Comfort as Affected by Solar Radiation, by G. A. Hendrickson and J. H. Walker, *Heating and Ventilating*, November, 1932, and The Determination of Sun Effect on Summer Cooling Loads, by G. A. Hendrickson and J. H. Walker, *Heating and Ventilating*, June, 1933.

[6]Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, February, 1934).

transmission through the glass caused by the difference between the temperatures of the inside and outside air. Another reason for neglecting this glass transmission load is that the curves in Fig. 1 were based on the maximum intensity of solar radiation observed at the A.S.H.V.E. Laboratory during a three-year study, so results based on these curves will be amply high. It will be noted that Table 4 gives the amount of heat delivered through the window as 97 per cent of the solar radiation, which is greater than is indicated by the figures for absorption in the preceding paragraph. The explanation is that much of the radiation absorbed by the glass is delivered to the room.

Fig. 1 shows that the maximum solar intensity on any surface is of limited duration. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. The cooling load due to solar radiation therefore does not have to be figured as a steady load. Another point which should be noted is that the maximum solar radiation load on an east wall occurs early in the morning when the outside temperature is low.

In a recent paper by the A.S.H.V.E. Research Laboratory[7] it was shown that ordinary double strength window glass transmits no measurable amount of energy radiated from a source at 500 F or lower; that it transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively; and that it transmits 65.7 per cent of the radiation from an arc lamp, 76.3 per cent of the radiation from an incandescent tungsten lamp, and 89.9 per cent of the radiation from the sun. Thus, glass windows in a room constitute heat traps, which allow rather free transmission of radiant energy into the room from the sun to warm objects in it, but do not allow the transmission of re-radiated heat from these same objects.

Some recent tests[4] indicated that sunshine through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different times.

Heat and Moisture Leakage

An allowance must be made for the heat and moisture in the outside air introduced for ventilating purposes or entering the building through cracks, crevices, doors, and other places where infiltration might occur.

[7]Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1933).

The volume of air entering due to infiltration may be estimated from data given in Chapter 6, and information on the amount of outside air required for ventilation will be found in Chapter 2.

The heat gain resulting from the outside air introduced may be estimated from the following formula:

$$H_i = Qd_o\,(\Theta_o - \Theta) \tag{2}$$

where

H_i = heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering the building, cubic feet per hour.
d_o = density of outside air, pounds of dry air per cubic foot of outside air, at the temperature t_o.
Θ_o = heat content of mixture of outside dry air (at temperature t_o) and water vapor, Btu per pound of dry air.
Θ = heat content of mixture of inside dry air (at temperature t) and water vapor, Btu per pound of dry air.

Heat and Moisture Sources

Figs. 6 to 9, Chapter 2, show the heat and moisture given off by human beings under various conditions of activity. For average conditions where a person is normally at rest, as in a theater, or doing very light work, as in a restaurant or residence, the total amount of heat given off will average about 400 Btu per hour. Part of this is latent heat due to the evaporation of 700 to 1200 grains of moisture per hour. Examples illustrating heat and moisture loss calculations for human beings are given in Chapter 2.

TABLE 5. HEAT GAIN DUE TO VARIOUS DEVICES, BTU PER HOUR

Lights and electric appliances	3,415 per kilowatt
Motors, 1/10 hp	255
Motors, 1 hp	2,546
Restaurant coffee urns, 10-gal capacity	16,000
Dish warmers per 10 sq ft of shelf	6,000
Restaurant range—4 burners and oven	100,000
Residence gas range	
Giant burner	12,000
Medium burner	9,000
Oven	1,000 per cu ft of space
Pilot	250
Electric Range	
Small burner, 100 to 1350 watts	3,415 to 4,600
Large burner, 1700 to 2200 watts	5,800 to 7,500
Oven, 2000 to 3000 watts	6,830 to 10,245
Appliance connection, 660 watts	2,250
Warming compartment, 300 watts	1,025

All sources of heat must of course be considered in designing the conditioning system. The heat gain due to various devices is given in Table 5. An example of cooling load calculation is given in Chapter 9.

PROBLEMS IN PRACTICE

1 ● a. What should be the dry- and wet-bulb temperatures in a restaurant when the outdoor dry-bulb temperature is 95 F?

b. What is the most desirable indoor dry-bulb temperature and relative humidity in an office building in summer?

a. Dry-bulb, 80 F; wet-bulb, 65 F. (Table 2, Chapter 2.)

b. 76.5 F and 50 per cent relative humidity. (Fig. 3, Chapter 2.)

2 ● The outdoor and indoor temperatures are 90 F and 78 F, respectively. What is the amount of heat transmitted per hour through a 7 ft by 4 ft north window?

$H_t = 28 \times 1.13\ (90 - 78) = 380$ Btu per hour.

(Equation 1, Chapter 8 and Table 12, Chapter 5.)

3 ● What are the proper design temperatures for a Detroit store?

Outdoor dry-bulb, 88 F; wet-bulb, 72 F. (Table 1, Chapter 8.)

Indoor dry-bulb, 77.5 F; wet-bulb, 64.5 F. (Table 2, Chapter 2.)

4 ● a. What is the maximum heat transmission for a flat roof exposed to the sun with the outdoor and indoor temperature 95 F and 80 F, respectively? The roof is of uninsulated 6-in. concrete, with its underside exposed, and with a black upper surface.

b. If the temperatures specified were the maximum for the day and occured at 12 o'clock, at what time would the maximum cooling load due to the roof exist?

a. $H_t = 1 \times 0.64\ (95 + 45 - 80) = 38.4$ Btu per hour per square foot.

(Equation 1 and Table 2, Chapter 8, and Table 11, Chapter 5.)

b. At 3 p.m. (Table 3.)

5 ● For south windows equipped with canvas awnings, what is the maximum amount of heat delivered to a room when the outdoor temperature is 90 F and the indoor temperature is 78 F?

$115 \times 0.28 = 32.2$ Btu per square foot of glass (Fig. 1 and Table 4; note that glass transmission can be neglected).

6 ● What is the heat gain per cubic foot of outside air introduced, under the following conditions:

Outdoor temperatures, 90 F dry-bulb and 75 F wet-bulb.

Inside temperatures, 78 F dry-bulb and 65 F wet-bulb.

$H_i = Qd_o\ (\Theta_o - \Theta)$. Equation 2.

The relative humidity of the outdoor air is 50 per cent (Fig. 3, Chapter 2), and $d_o = \frac{1}{14.21} = 0.0703$ (Table 5, Chapter 1).

$\Theta = 37.81$ and $\Theta = 29.65$ (Table 5, Chapter 1). The total heat of any air-vapor mixture may be obtained from the last column in Table 5, Chapter 1, by considering the temperatures to be wet-bulb readings, since the total heat of a mixture is constant for a given wet-bulb temperature.

$H_i = 1 \times 0.0703\ (37.81 - 29.65) = 0.57$ Btu per cu ft.

7 ● If there are twenty 200-watt lights in use in a room, what is the cooling load due to lights?

$200 \times 20 = 4000$ watts $= 4$ kw.

$3415 \times 4 = 13{,}660$ Btu per hour (Table 5, Chapter 8).

8 ● a. When a restaurant has two 10-gal coffee urns, what is the cooling load due to them?

b. What is the cooling load due to four 1350-watt burners on an electric range?

a. $16{,}000 \times 2 = 32{,}000$ Btu per hour (Table 5, Chapter 8).

b. $4600 \times 4 = 18{,}400$ Btu per hour (Table 5, Chapter 8).

Chapter 9

CENTRAL AIR CONDITIONING SYSTEMS

Types of Systems, Dehumidifiers, Designing the System, Zoning, Location of Apparatus, Temperature of the Air Leaving Outlets, Air Quantity Required, Heat to be Removed by Cooling and Dehumidifying Apparatus, Size of Reheaters, Surface Cooling Problems, Auxiliary Equipment

CENTRAL systems, equipped for cooling and dehumidifying, are used principally in the air conditioning of theaters, restaurants, office buildings, or other places where many people gather, and in manufacturing establishments where air conditions have an important influence on the quality of product or rate of production. A central cooling and dehumidifying plant is one in which the fans, dehumidifiers, and other related apparatus are assembled in suitable apparatus rooms from which distribution and return ducts lead to the conditioned spaces. The design of such systems is considered in this chapter, while in Chapter 22 central systems for heating and humidifying are described. Industrial air conditioning has been considered in Chapter 3.

TYPES OF SYSTEMS

Dehumidification or cooling of air may be accomplished by several methods and by use of many heat transfer mediums. Most comfort-conditioning, central station, air-conditioning systems employ cold water or the direct expansion of a refrigerant in either spray type or surface type equipment to accomplish the required cooling and dehumidification. Among the several other methods that may be employed are: passing the air through or over a dehydrating agent and then lowering the dry-bulb temperature to the proper level, and evaporative cooling. The former method is applicable to comfort conditioning only where reasonably cold water is available for reducing the dry-bulb temperature after dehydration, while the latter method is applicable to comfort conditioning only in regions where the summer wet-bulb temperature is low.

If the system is intended solely for summer conditioning, the apparatus will consist essentially of a dehumidifier of the surface type or spray type; filters; fan and motor; reheater; outside air, return air, and supply air duct work; air outlets and grilles; spray pump for spray dehumidifier; refrigeration equipment; and suitable controls. Generally, however, a central station air conditioning system is designed for year-round service. This means that properly sized heaters and humidifiers, with their respective

controls, must be added. With few exceptions, systems designed to meet summer capacity requirements will have ample capacity for winter and intermediate season conditioning.

A common arrangement of a central station spray type system for cooling and dehumidifying is illustrated in Fig. 1. The plant may be designed to condition 100 per cent outside air, 100 per cent return air, or a mixture of outside and return air. Further, part of the air returned from the conditioned space may be by-passed[1] around the conditioner as illustrated in Fig. 2. The reheater may be installed in the fan inlet chamber as shown, in the by-pass air duct, or in the fan discharge duct, depending upon apparatus space and other design conditions. Still another arrangement of equipment will result if the dehumidified air fan delivers the conditioned air to several other fans rather than to the conditioned space directly.

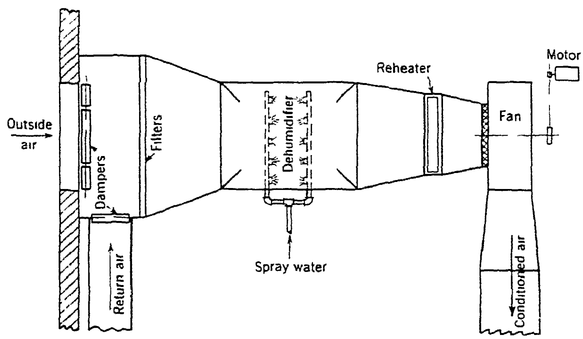

FIG. 1. SPRAY TYPE AIR CONDITIONING APPARATUS

These booster fan equipments may use part by-pass air as illustrated in Fig. 3 or 100 per cent dehumidified air and reheaters. The main apparatus, in either case, may or may not have a by-pass connection, depending on load conditions and other design factors.

The systems illustrated in Figs. 1 and 2 may be converted into the surface cooling type by merely replacing the dehumidifiers with surface cooling coils which use cold water or direct expansion of refrigerant to accomplish the required cooling and dehumidifying. The coils may also be installed within the spray chamber, either in series with the sprays or below them.

DEHUMIDIFIERS

Information on spray type dehumidifiers is given in Chapter 11.

Surface cooling type dehumidifiers generally consist of extended-surface coils within which the water or refrigerant is circulated or the refrigerant is expanded. The air to be cooled and dehumidified is drawn or blown over the coils. This system is generally comparatively low in initial cost and has low operating costs. For comfort cooling, water is usually used to

[1]Patents exist covering the use of the by-pass for cooling and dehumidifying systems.

bring the refrigeration effect to the coils. Many localities have refrigeration codes which restrict the use, in comfort conditioning applications, of refrigerants acting by direct expansion in coils exposed to the air stream. Therefore, local codes should be consulted by the designer before he plans a system employing direct-expansion methods. Close humidity control cannot be maintained during the cooling season by the surface cooling type of equipment. Winter humidification may be accomplished by use of evaporating pans or spray nozzles. The cooling coils serve no purpose during the intermediate or heating seasons, so in this respect the spray type equipment is often preferred, in that during certain seasons evaporative cooling will be sufficient to produce the cooling desired. Effective cooling and dehumidification accomplished by surface units are dependent upon many variable factors. The air velocity through the unit, air

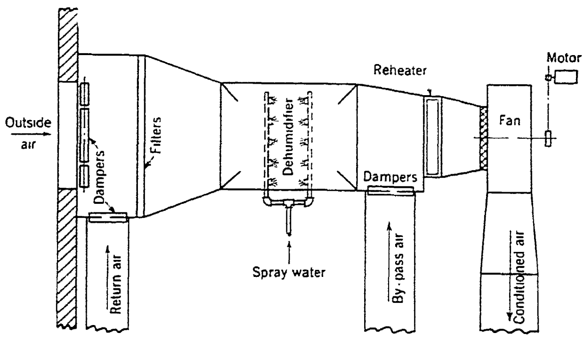

FIG. 2. SPRAY TYPE AIR CONDITIONING APPARATUS WITH BY-PASS

temperature, moisture content of the air, water or refrigerant temperature, and velocity of the water or refrigerant through the tubes must be considered in selecting the proper unit for a given design load. If any of these factors vary without a corresponding variation of the other factors, the effective work of the coil will increase or decrease, as the case may be.

DESIGNING THE SYSTEM

The general procedure for the design of a central cooling and dehumidifying system is as follows:

1. Calculate the heat gain for each room or space to be conditioned. (See Chapters 5 and 8.)
2. Determine the volume of outside air to be introduced. (See Chapter 2.)
3. Assume or calculate the temperature of air leaving the supply outlets.
4. Calculate the quantity of air to be circulated.
5. Estimate the temperature loss in the duct system.
6. Calculate the heat to be removed by the cooling and dehumidifying apparatus.
7. Calculate the size of the reheating equipment.
8. Select cooling equipment and heating equipment from manufacturers' data and performance curves.
9. Calculate total tonnage.

10. Design the air distribution system and the air outlets and inlets. (See Chapters 19 and 20.)
11. Calculate the total static pressure of the system.
12. Select the fan, motor, and drive. (See Chapter 17.)
13. Select the pump and motor.
14. Design the control system. (See Chapter 14.)

ZONING

The above general outline of procedure will prove satisfactory for the smaller and less complex installations. However, when dealing with air-conditioning systems for large buildings, after a proper analysis has been made of the conditions to be maintained and the heat loads encountered, it is generally considered best practice to divide the complete job into a

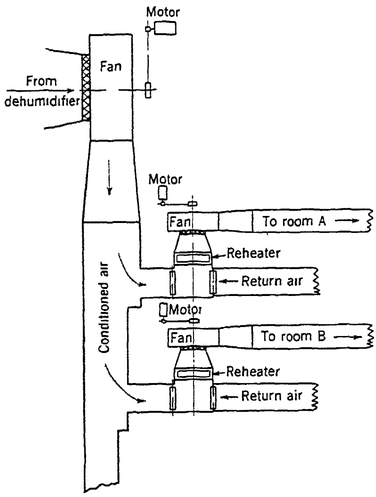

FIG. 3. CENTRAL DEHUMIDIFYING PLANT AND LOCAL RECIRCULATING FANS

number of suitably sized units. In some cases a unit per floor or group of floors may complete the design satisfactorily, whereas in others it may be advantageous to have separate units for each of the various outside exposures of the building. Where the floor area is large in relation to the outside wall exposure, it is obvious that provision must be made for the variable load to which the outside exposures are subjected. The heat loads on inside rooms are apt to be less variable since the fluctuations of the outside weather conditions are not directly involved. Such conditions often result in the natural zoning or segregation of rooms having similar exposures and internal heat loads.

LOCATION OF APPARATUS

Availability of space for apparatus and duct work is of primary importance when selecting the type of system for a given design. In general, for large installations, the refrigeration equipment, because of its size,

weight, and operating characteristics, is located in the basement along with the boilers, fire pumps, and other equipment. The air conditioning apparatus is generally located where clean outdoor air is readily available, the designer bearing in mind that supply and return air ducts, steam connections, water and drain connections, and electrical connections must be made to the equipment proper.

TEMPERATURE OF AIR LEAVING OUTLETS

In comfort conditioning applications, air has been distributed from properly designed outlets without producing drafts at temperatures varying from approximately five to thirty degrees below the required room temperature. Factors influencing the design and selection of air outlets are: ceiling height, type of ceiling, length of blow, and temperature and quantity of air to be distributed. Most summer conditioning installations are designed to supply the air to the conditioned space at from 8 to 15 deg below room temperature. Recently the use of specially designed nozzles has indicated the possibility of reducing the air quantity necessary to dissipate a given heat load by introducing the air into the room as much as thirty degrees below room temperature. Comfort conditioning systems employing differentials greater than fifteen degrees require special consideration and design experience because high pressure outlets or nozzles are usually used. Further, care must be taken to allow a sufficient air quantity under all load conditions to insure good distribution. If winter heating, as well as summer conditioning, is to be accomplished by the same distributing system, the design of the outlets will be influenced as discussed in Chapter 22. Industrial systems in which drafts are not objectionable usually employ a temperature differential equal to the dew-point depression.

AIR QUANTITY REQUIRED

For calculating the quantity of air required to absorb a given heat gain, the following approximate formulae may be used:

$$M = \frac{H_s}{60 \times 0.24 \times (t - t_y)} = dQ \qquad (1)$$

or, assuming a constant value of 0.075 lb for d,

$$Q = \frac{H_s \times 55.2}{60 \times (t - t_y)} \qquad (2)$$

where

Q = volume of air required, cubic feet per minute.
H_s = total sensible heat gain, Btu per hour.
t = room temperature, degrees Fahrenheit.
t_y = outlet temperature, degrees Fahrenheit.
M = weight of air required, pounds per minute.
d = density of air at the temperature and relative humidity of the room, pounds per cubic foot.

Example 1. The total sensible heat gain in a restaurant when held at 80 F is 199,736 Btu per hour. Assuming a 12 deg Fahr temperature differential between the entering air and the room temperatures, which is the same as assuming the dry-bulb temperature of the entering air to be 68 F, calculate the required air capacity of the system.

Solution.

$$Q = \frac{199{,}736 \times 55.2}{60 \times 12} = 15{,}313 \text{ cfm} = 1146 \text{ lb per minute.}$$

If a system similar to the one shown in Fig. 1 is used, 1146 lb per minute will be the capacity of the dehumidifier as well as of the fan equipment.

Example 2. If in addition to the 199,736 Btu per hour sensible heat load, the conditioned space has a moisture gain of 384,000 grains per hour, calculate the apparatus dew point required to give maintained conditions of 80 F dry-bulb and 65 F wet-bulb, with a corresponding 56½ F dew point.

Solution. With 384,000 grains of moisture per hour to be picked up, the entering dew-point temperature should be low enough so that the addition of this moisture will not increase the dew point above 56½ F.

Grains per pound of air saturated at 56½ F (Table 5, Chapter 1)		68.1
Less: Grains per pound to be picked up,	$\frac{384{,}000}{1146 \times 60}$,	5.6
Grains per pound allowable in entering air		62.5

This corresponds to an apparatus dew-point temperature of 54.17 F.

Example 3. Illustration of the by-pass system. (See Fig. 2.)

Assume the same data as for Example 2. Instead of passing all of the air through the dehumidifier for cooling and dehumidifying, a portion may be passed through and the balance be mixed with the conditioned air at the leaving end of the dehumidifier, the mixture being proportioned so that the resultant conditions will be those required to give proper conditions in the area considered.

Solution. The quantity of air to be dehumidified, the quantity to be by-passed, and the apparatus dew-point temperature may be calculated as follows:

Let

X = percentage of air to be by-passed.
Y = percentage of air to be passed through the dehumidifier.
t_d = apparatus dew-point temperature, degrees Fahrenheit.

The quantity X of 80-F air must mix with the quantity Y of dehumidified air to produce air with a resultant 65 F wet-bulb temperature. Also, X quantity of air at 56½ F dew point must be mixed with Y quantity of dehumidified air to give a resultant apparatus dew-point temperature of 54.17 F. It is assumed that the air passing through the dehumidifier is saturated.

Solving simultaneous equations,

$$80.0X + Yt_d = 68.00 \qquad (3)$$

$$56.5X + Yt_d = 54.17 \qquad (4)$$

$$23.5X + 0 = 13.83$$

$$X = \frac{13.83 \times 100}{23.5} = 59 \text{ per cent, air by-passed.}$$

$$Y = 100 - X = 41 \text{ per cent, air passed through washer.}$$

The second step is to determine the apparatus dew-point temperature. Substitute X in either Equation 3 or Equation 4, and solve for t_d:

$$80 \times 0.59 + t_d \times 0.41 = 68$$

$$t_d = \frac{68 - 47}{0.41} = 51.2 \text{ F, the apparatus dew point.}$$

HEAT TO BE REMOVED BY COOLING AND DEHUMIDIFYING APPARATUS

Example 4. Assume the same data as for Example 3. If the amount of outside air, at 95 F dry-bulb and 75 F wet-bulb, required for ventilation has been found to be 169 lb per minute, determine the refrigeration capacity required.

Solution. As the total weight of the air introduced per minute is 1146 lb, and 41 per cent of it goes through the dehumidifier, the total work to be done may be computed as follows:

Air passing through humidifier, 1146 × 0.41	470 lb
Less: Outside air for ventilation	169 lb
Return air	301 lb

The refrigeration required for the return air is:

Total heat per pound at 65 F	29.65 Btu
Less: Total heat per pound at 51.2 F	20.85 Btu
Requirement for cooling 1 lb of return air	8.80 Btu

301 lb × 8.80 Btu = 2649 Btu per minute required to cool the return air.

The refrigeration required for the outside air is:

Total heat per pound of outside air	37.81 Btu
Less: Total heat per pound at 51.2 F	20.85 Btu
Requirement to cool 1 lb of outside air	16.96 Btu

169 lb × 16.96 Btu = 2866 Btu per minute required to cool the outside air.

Thus, the total refrigeration required is:

2649 Btu + 2866 Btu = 5515 Btu per minute, which is equivalent to a load of 27.6 tons of refrigeration.

SIZE OF REHEATERS

A properly designed air-conditioning system will have reheaters of sufficient capacity to heat the conditioned air from the apparatus dew-point temperature to the outlet delivery temperature. If winter heating is to be accomplished, consult Chapter 22.

The following general formula may be used to determine the amount of heat necessary to reheat a given quantity of air:

$$H_l = 0.24\ (t_y - t_d)\ M \qquad (5)$$

where

H_l = heat to be supplied to reheater coil, Btu per hour.

Example 5. Assume the same data as for Example 1, and find the amount of reheating required.

Solution.

$$H_l = 0.24\ (68 - 54.17)\ 1146 \times 60 = 228{,}200 \text{ Btu per hour.}$$

SURFACE COOLING PROBLEM

The amount of coil surface required for a given amount of work is dependent upon factors previously listed. Obviously, the various types of surfaces made available by different manufacturers will have different

transmission values. It is recommended that the designer consult the latest manufacturers' catalogs because more accurate ratings are being issued from time to time.

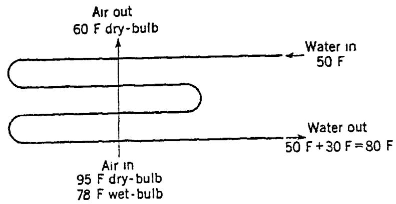

FIG. 4. COUNTER-FLOW SURFACE COOLING DIAGRAM

Example 6. It is desired to cool and dehumidify 30,000 cfm of air at 95 F dry-bulb, 78 F wet-bulb, and 72 F dew point, to a 60 F dew point. Cooling water is available at 50 F in a quantity which will allow a 30 F rise in temperature to be used. The counter-flow surface cooling used is sketched in Fig. 4.

Solution. The pounds of partially saturated air cooled and dehumidified per hour equal 60 times the cubic feet of air at 95 F dry-bulb and 78 F wet-bulb brought past the coil surface per minute, multiplied by the pounds per cubic foot of the air as determined from Table 3, Chapter 1.

$$30{,}000 \times 60 \times 0.0708 = 127{,}440 \text{ lb per hour.}$$

The total heat H_t to be removed per hour by the surface coil is found to be equal to the pounds of partially saturated air passed over the coil per hour times the difference between the total heat of air at 78 F wet-bulb and at 60 F wet-bulb.

$$H_t = 127{,}440\ (40.64 - 26.18) = 1{,}842{,}000 \text{ Btu per hour.}$$

The latent heat H_l to be removed per hour will be found by multiplying the pounds of partially saturated air passed over the coils per hour by the difference in the latent heat of the air per pound at the initial and final dew points.

$$H_l = 127{,}440\ (17.79 - 11.69) = 777{,}000 \text{ Btu per hour.}$$

The sensible heat H_s to be removed per hour is equal to the total heat of the air less its latent heat.

$$H_s = H_t - H_l = 1{,}842{,}000 - 777{,}000 = 1{,}065{,}000 \text{ Btu per hour.}$$

Manufacturers' standard ratings for surface coolers are usually based on the cubic feet of air passed through their equipment per minute, reduced to the conditions of saturated air measured at a temperature of 70 F. In the present example, to convert the 127,440 lb of air cooled per hour to a basis which will permit the use of such standard ratings, it is necessary to multiply the pounds of air cooled per hour by the specific volume of the air, and to divide by 60.

$$\frac{127{,}440 \times 13.69}{60} = 29{,}100 \text{ cfm of 70 F saturated air.}$$

The amount of cooling water necessary when a 30 degree rise in its temperature is to be used is:

$$\frac{1{,}842{,}000}{30 \times 8.34 \times 60} = 123 \text{ gpm.}$$

With counter flow of air and water, it is necessary to determine the mean temperature difference between the air and the water in order to properly use the transmission coefficients given in apparatus rating tables.

$$\text{Mean temperature difference} = \frac{D_1 - D_2}{\log_e \frac{D_1}{D_2}} \quad (6)$$

where

D_1 = the difference between the temperatures of inlet air and outlet water, degrees Fahrenheit.

D_2 = the difference between the temperatures of outlet air and inlet water, degrees Fahrenheit.

$$\frac{(95 - 80) - (60 - 50)}{\log_e \frac{(95 - 80)}{(60 - 50)}} = 12.33 \text{ F.}$$

If from apparatus rating tables based on air velocities over the coils and water velocities through the coils, it has been found that the transmission coefficient is equal to 8.0 Btu per square foot per degree difference in mean temperature between the air and the water, the area of cooling coil surface necessary will be equal to the sensible heat divided by the transmission coefficient and also by the mean temperature difference.

$$\frac{1{,}065{,}000}{8.0 \times 12.33} = 10{,}800 \text{ square feet of cooling coil surface necessary.}$$

The latent heat is taken out at the same time the sensible heat is extracted, but no extra surface is required unless the latent heat exceeds approximately 40 per cent of the total heat. This is because the wetted surface has a much higher coefficient of transmission. Approximately 10 per cent more surface should be added if the latent heat exceeds 40 per cent of the total heat.

AUXILIARY EQUIPMENT

Consult Chapters 14, 17, 19, 20, and 22 for information on the air distribution system; air outlets and inlets; static pressure on fan; fan motor, and drive; and the control system.

PROBLEMS IN PRACTICE

1 ● In summer air conditioning what factors control the difference between the dry-bulb temperature of the conditioned space and the dry-bulb temperature of the entering air?

1. The duct and supply grille arrangement permitted by architectural and structural requirements for the particular space, *e.g.*, ceiling height and obstructions on ceilings, such as beams.
2. The state of activity of the occupants.
3. The outlet velocity at the grille, as limited by noise level requirements.
4. The direction of the jet relative to the occupants.
5. In some cases, the temperature of the available water supply, which may have some bearing on the air delivery temperature.

2 ● What factors determine the volume of conditioned air which must be delivered to the space?

The sensible heat to be removed, and the allowable temperature differential.

3 ● What factors determine the dew point of the air entering the space?

The maximum dew point desired in the conditioned space, and the moisture gain in the space per unit weight of air supplied.

4 ● Why must the air leaving a dehumidifying type air washer be reheated before delivery?

The air leaves the dehumidifying air washer saturated at a relatively low temperature which in most cases is lower than the allowable delivery dry-bulb temperature. Also, the air may possibly be carrying a small amount of entrained water which might settle out in the ducts near the washer and cause corrosion difficulties.

5 ● What methods are used for reheating air?

1. Passing it over reheating coils.
2. Mixing it with by-passed air at a higher temperature.

6 ● What determines the final temperature of the spray water in a dehumidifier?

Because of the effectiveness of the heat transfer between air and finely divided spray water in a well designed dehumidifier, the air will be cooled to within 1 or 2 F of the final water temperature, provided the air velocity through the washer does not exceed 600 fpm. This final temperature should then be taken as 1 or 2 F lower than the required dew point of the air leaving the washer.

7 ● What are the advantages of using counter flow of air and water in surface coolers?

Counter flow results in a higher mean temperature difference than does parallel flow for the same range of air and water temperatures, which means that less cooling surface is required. Counter flow permits higher initial water temperatures and also allows a greater temperature rise for the water. These factors combine to reduce the cost of circulating and refrigerating the cooling water.

8 ● What factors other than cost should be considered in determining whether to use a central system or another type?

a. Appearance: The equipment must be designed to harmonize with the architecture of the building.

b. Distribution: The system must maintain adequate and uniform air motion over the entire conditioned space.

c. Control: The control system must be designed to give effective partial load operation.

9 ● Can the central cooling and dehumidifying system be used as an all-year-round conditioner?

By modifying the control system and adding blast coils or a water heater to the spray type system, the cooling system will function as one for heating and humidifying. The surface cooling type may be transformed by modifying the control, and adding another set of coils and a humidifier.

10 ● Will the tons of refrigeration-effect per day be the value calculated in Example 4 of this chapter times the hours of operation?

No. The tons of refrigeration-effect are functions of the load. The components of the load vary, that is, the number of people occupying the space, the outdoor conditions, and the solar radiation will change from hour to hour and from day to day. The calculated load represents the maximum required for design peak conditions.

11 ● Will the quantity of return air required in Example 4 of this chapter be used all season?

No. When the outdoor wet-bulb temperature becomes lower than the maintained wet-bulb temperature, it is more economical to use all outside air than to dehumidify the return air.

Chapter 10

COOLING METHODS

Methods of Cooling Air, Evaporative Cooling, Dehumidification, Silica Gel System, Alumina System, Design of System, Operating Methods, Steam Jet System, Compressors, Refrigerants, Methods of Cooling, Condensers

BY using any of the following four methods, or any combination of them, *effective temperature* (see Chapter 2) may be reduced.

a. Sensible cooling: Lowering of the dry-bulb temperature by the removal of sensible heat without change of the dew-point temperature.

b. Dehumidifying: Lowering of the dew-point temperature by the removal of moisture without change of the dry-bulb temperature.

c. Evaporative cooling: Lowering of the dry-bulb temperature through the evaporation of moisture without the addition or the subtraction of heat.

d. Air motion: Increasing the air motion over the body with the resulting higher evaporation from the skin.

As an example, let the condition be considered of 92 F dry-bulb, with a 40 per cent relative humidity, corresponding to a wet-bulb temperature of 72.8 F, and an effective temperature for still air of 81.1 F. This *effective temperature* may be reduced 3.1 F by any of the four basic methods mentioned, as follows:

First, by lowering the dry-bulb temperature to 85.5 F without changing the dew-point of 64.2; this gives an effective temperature of 78 F.

Second, by reducing the moisture content of the air to 46 grains per pound of dry air without changing the dry-bulb temperature; this gives an effective temperature of 78 F.

Third, by reducing the dry-bulb temperature to 83.8 F without changing the total heat of the air. This requires the evaporation of 14 grains of moisture per pound of dry air, and the effective temperature will become 78 F.

Fourth, by increasing the air movement from still air to 460 fpm, a velocity which will reduce the effective temperature 3.1 F from 81.1 F to 78 F.

Method to Employ

The best method of reducing the effective temperature in any specific case will depend on the accompanying circumstances and can be determined only by a thorough analysis made by a competent engineer. Generally speaking, the removal from the air of the sensible heat, or moisture, or both, by sensible cooling or dehumidifying is the most satisfactory method. Adequate results by the utilization of air motion or by evaporative cooling are difficult to obtain because of the dependence of both methods upon climatic conditions beyond the engineers' control although these methods are much less expensive than the first two

mentioned. Cooling by evaporation is satisfactory only when the air to be cooled is very dry; air motion as a means of producing cooling effect is never entirely adequate in the range of high temperatures. Of the two, evaporative cooling, or adiabatic saturation of the air, is a much more dependable method which will make more reduction in the effective temperature than will an increasing air motion within permissible limits.

As an example of this, consider an outdoor condition of 96 F dry-bulb and 80 F wet-bulb. The effective temperature is 85.7 F and, if the still air is moved with a velocity of 300 fpm, the effective temperature will be reduced only 2.0 F while saturation at the wet-bulb temperature would reduce the effective temperature 5.7 F. At 300 fpm velocity this saturated air would reduce the effective temperature to 75.6 F, thus making a total improvement of 10.1 F.

Evaporative Cooling

Evaporative cooling is accomplished by passing air through a water spray in which the water is being continually recirculated. The air, entering in an unsaturated condition, evaporates a part of the water at the expense of the sensible heat. As this is an adiabatic transfer, the total heat content of the air remains constant, while the dew point rises and the dry-bulb falls until the air is saturated. A system[1] of ducts and a propelling fan are used to distribute the air in a proper manner.

It will be seen that the reduction in dry-bulb temperature is a direct function of the wet-bulb depression of the air entering the dehumidifier and that the resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air.

Dehumidification

Dehumidification may be accomplished in three ways:

1. By cooling the air below the dew point and causing a part of the moisture contained to precipitate.
2. By extracting all or part of the moisture by absorption.
3. By extracting all or part of the moisture by adsorption.

As used in this discussion, the term *adsorption* pertains to the action of a substance in condensing a gas or vapor and holding the condensate on its surface without any change in the chemical or physical structure of the substance and with the release of sensible heat. The term, *absorption*, implies a change in the chemical or physical structure of a substance in the process of dehydrating air. Adsorbers include silica gel and lamisilite; absorbers include sulphuric acid.

Dehumidification by Refrigeration

Air conditioning imposes requirements on refrigeration equipment not usually found in general cooling work, so that specially designed apparatus is often needed to replace that normally used for industrial cooling. Standard equipment can be adapted to meet air conditioning requirements but extreme care must be taken to determine the limits of its applicability.

[1]See Air Washer Performance in Chapter 11; also Theory of Atmospheric Cooling in same chapter.

In industrial or process cooling systems the load is fairly constant, noise in operation is not of paramount importance, space is available or relatively cheap, condenser water is not a source of worry, and the cooling system is to a great extent separate and independent of other mechanical equipment. By contrast, air conditioning, especially as used for space cooling and comfort work in office buildings, theaters, and places where people gather requires special consideration of all these factors. Space in public buildings is limited and condenser water is expensive. Noise interferes with the occupants, and the cooling equipment must dovetail with the other air-handling apparatus. Most important, the load fluctuates tremendously and is seasonal.

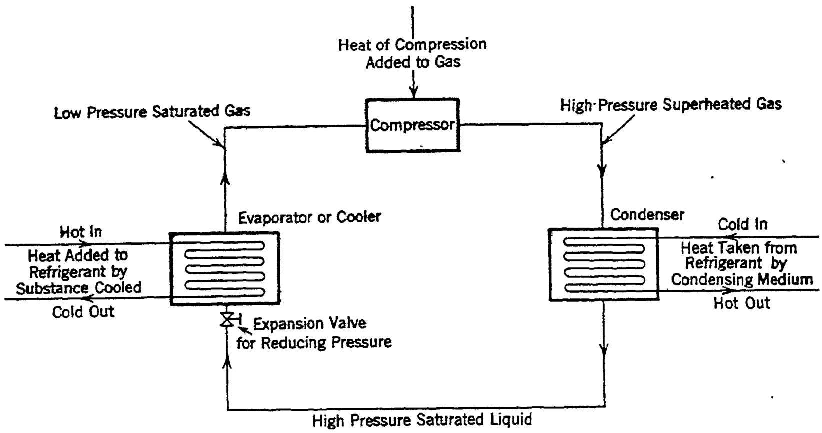

FIG. 1. TYPICAL REFRIGERATION DIAGRAM

A complete discussion of the thermodynamic problems of refrigeration is given in the Refrigerating Data Book[2], 1934, so only a brief description of the cycle will be given here before the problems peculiar to air conditioning are considered.

The refrigeration system consists of three main parts, the evaporator, the condenser, and the compressor. Fig. 1 shows a diagram of the cycle. Heat is absorbed in the evaporator and released in the condenser. The compressor changes the level of the heat by taking it from a lower to a higher plane. There are also many valves, accessories, and special devices necessary for proper operation, which vary somewhat with different types of cooling systems and different refrigerants.

In a simple illustrative cycle of a refrigeration system, the liquid refrigerant under high pressure has both its pressure and temperature reduced by being expanded through a suitable valve into an evaporator or cooler. Within the evaporator the low temperature of the refrigerant allows it to absorb heat from the substance to be cooled, which surrounds the evaporator. This absorption of heat increases the pressure of the

[2]Published by *American Society of Refrigerating Engineers.*

refrigerant, and a compressor is employed to withdraw enough low-pressure saturated gas to keep the cooling action of the evaporator continuous. The withdrawn gas is discharged from the compressor to the condenser in the form of a high-pressure superheated gas which includes the heat added through its compression. In the condenser, because heat is taken from the gaseous refrigerant by the condensing medium, usually water, the refrigerant again becomes the high-pressure saturated liquid with which the cycle started.

The cooling water, which may come from a deep well or from a city main, may be utilized for some purpose after it has been warmed a few degrees in the condenser, or after use it may be exposed to the atmosphere

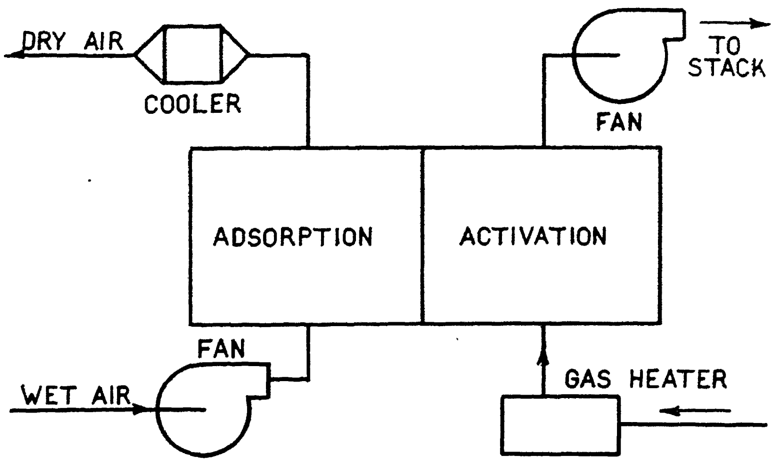

FIG. 2. SILICA GEL AIR-CONDITIONING SYSTEM—SINGLE STAGE ADSORPTION

in a spray pond or cooling tower and have its temperature reduced to a point where the water may be used again. (See Chapter 11.)

Silica Gel System

Silica gel is a chemical composition made from sodium silicate and acid, the chemical formula being SiO_2. It has an appearance greatly resembling that of clear quartz sand but it differs in structure in that the crystals are highly porous, with voids constituting 41 per cent by volume although the pores are microscopic in size. This material possesses the property of being able to adsorb a substantial portion (about 25 per cent of its own weight) of moisture from the air without any increase in its volume. After the silica gel has become "saturated" or has adsorbed moisture to the limit of its capacity, the moisture may be driven from it by the application of heat, again without change in the structure, volume, or chemical composition of the silica gel. This cycle may be repeated indefinitely. When applied to air conditioning the silica gel which is exposed to the air reduces the moisture content in the air and releases sensible heat which may be readily removed from the air. A typical diagram is shown in Fig. 2.

Practical Application of Silica Gel

Silica gel has two applications when used to replace refrigeration. In the one principally used, the air from which moisture is to be extracted is taken through silica gel beds by suction or pressure fans, and by means of this process the moisture becomes adsorbed by the silica gel and the air leaves at a lower dew point and a higher sensible temperature than those at which it entered. If this air is passed over surface coolers in which tap water or another cooling medium is flowing through tubes, a certain amount of sensible heat will be removed. The air leaves the surface cooler or interchanger with the same dew point with which it emerged from the silica gel beds, but with a lower dry-bulb temperature, although the dry-bulb temperature may be higher than the temperature of the air entering the silica gel beds.

In another method, the first two of the steps outlined are duplicated, and in addition the air is carried through a spray type washer. Because the air enters the washer with a low wet-bulb, and because adiabatic saturation will take place at a temperature close to the entering wet-bulb, considerable cooling of the air can be accomplished; but this can be done only with a consequent increase of the dew point.

It is necessary to reactivate the silica gel after it has adsorbed about 25 per cent of its own weight in the form of moisture. As reactivation requires a high temperature and since silica gel is only active at low temperatures, cooling of the beds must also be completed before they can be used again. This necessitates three stages in the silica gel containers and requires either three beds of silica gel or one bed divided and automatically put in position. The reactivation is usually done by means of gas or oil fires and the cooling of the beds by means of indirect water cooling or by means of small quantities of dehydrated air taken from the system beyond the interchanger.

Alumina System of Adsorption

Activated alumina contains a trifle over 91 per cent of aluminum oxide, Al_2O_2, which material will adsorb nearly 100 per cent of the vapor in the air up to about 8 or 10 per cent of the weight of the adsorbing material, after which the adsorption falls off gradually as the saturation point is approached. The application is quite similar to that employed for silica gel; that is, the material is exposed to the air flow and after reaching about 75 per cent saturation is reactivated by removing the moisture adsorbed by means of applied heat. The actual scheme generally followed in the use of this material for continuous service varies somewhat from silica gel inasmuch as the material is placed in three units which are used consecutively for the different steps. These steps permit each unit to operate as follows:

a. In series with the preceding unit.
b. Alone.
c. In series with the following unit.

This plan allows for adsorption, reactivation, and cooling, in a manner similar to that used with silica gel.

Taking a single unit, when it is in the *a* step and operating with the

preceding unit, the alumina adsorbs approximately 25 per cent of the moisture in the air and takes up about 1.3 per cent of its weight of water. During the second step when it is operating alone, it takes up 100 per cent of the moisture in the air until the weight of the water adsorbed is brought up to about 6.7 per cent. During the third step when the unit is operating with the succeeding unit, it extracts about 75 per cent of the moisture in the air until the water weight adsorbed comes up to about 10 per cent of the weight of the adsorber. The time allowable for reactivating is equal to the time occupied by the second unit adsorbing alone, plus the time when the second and third units are adsorbing in series, plus the time when the third unit is adsorbing alone, at the expiration of which time the first unit will be again required.

The temperature of air used for alumina reactivation is usually between 300 and 700 F and the air flow rate will have to be higher with the low temperature air than it will be with reactivating air of higher temperature. For example, air at 400 F for reactivating will, at 10 cu ft per hour per pound of alumina, require about 6 hours for reactivation. In the three unit system, after reactivation the cooling of the activated alumina may be carried out with considerable rapidity by using dry air from the adsorption unit for circulation through the unit which has just completed reactivation. The final temperature of the unit before it goes back into service should be not over 200 F. As a basis for computing the amount of cooling air required for reactivation, each cubic foot of cooling air has been found capable of removing 2.2 Btu when heated from 85 to 200 F and of providing a sufficient margin of safety in operation.

Design of System

When designing air conditioning systems, the capacity of equipment is decided by selecting apparatus of sufficient size to maintain predetermined temperatures and humidities in treated spaces when arbitrarily established maximum atmospheric temperatures occur coincident with given conditions of population, lighting, and power consumption. These factors determine the maximum duty of the cooling system. The duty does not necessarily determine the size or capacity of the refrigeration apparatus. The refrigerating capacity is expressed in *tons*, each ton being equal to the absorption of the heat given up by one ton of ice at 32 F melting to water at 32 F in 24 hours. This is equivalent to heat absorption at a rate of approximately 200 Btu per minute, or 12,000 Btu per hour.

After the maximum duty is determined, the other factors concerning the installation must be investigated. The total heat to be removed by the cooling system has many sources, some substantially constant and others extremely variable. These sources can be roughly classified as follows, the first column indicating the order in amount and the second the order in variability:

1. Fresh air supplied.	1. Fresh air supplied.
2. Population.	2. Transmission through the structure.
3. Transmission through the structure.	3. Light and power consumed.
4. Light and power consumed.	4. Population.

By combining these two columns, a third grouping is obtained as follows:

1. Fresh air supplied.
2. Transmission through the structure.
3. Population.
4. Light and power consumed.

In this last arrangement, the first two items are governed by atmospheric conditions and they are therefore subject to tremendous fluctuations in value. As they generally form 40 to 60 per cent of the entire maximum load, the duty of the cooling system will be much less than maximum most of the time.

The transmission through the structure is especially influenced by the sun. (See Chapter 8.) In many cases, because of the heat flow resistance of the structure, the heat from the sun is retarded until it is compensated for by a reduced general temperature out-of-doors.

A survey of Weather Bureau records indicates that maximum temperatures occur less than 5 per cent of the cooling period and also that the duration of peak conditions is never more than three or four hours.

Two factors control the size of the refrigeration system, the evaporator or suction temperature, and the condenser or head temperature. With the knowledge that the system will operate most of the time with a load of not over 60 per cent of maximum, and that maximum demands will occur infrequently and only for short periods, some provision must be made to insure economical operation under average conditions. This can be done by overloading the machine under extreme demands and basing the design on normal or average loads. Flexibility in arrangement can be provided in several ways.

Variations in load change the efficiency of any machine and a refrigerating system can be costly and inefficient if improperly designed or operated. Fortunately, the trouble can be concentrated in the compressor and the problem relieved of many complications. It is comparatively easy to furnish condensers and evaporators to carry the maximum load so arranged that they will function properly at small demands. They affect the compressor performance to some extent but most of the compressor problems are in the machine itself.

Variations in load are usually effected by lowering the suction temperature and pumping a larger volume of gas per ton through a greater pressure range. This is possible because the latent heat of the refrigerant remains nearly constant throughout the small range used and the specific volume varies rapidly with change in pressure. As the compressor must remove the refrigerant evaporated, the evaporator temperature fixes the displacement required. The objection to such method is that the total power consumed remains nearly constant and the power per unit of cooling increases rapidly as the total output is reduced. Such operation is satisfactory as long as the load is kept within 10 per cent of the rating of the compressor but this condition does not commonly occur in air conditioning applications.

Operating Methods

It is possible to divide the entire refrigeration system into a number of small units, which will allow cutting in and out of compressors and condensers as the load fluctuates. This, however, is an expensive method as a number of small units are usually more expensive than one large unit. There is a certain amount of duplication of equipment necessary, which

tends to increase the initial cost of the system and which makes the fixed charges, applicable to the operation of the air conditioning and cooling system, greater than necessary.

A second method of providing for economy of operation is to have storage capacity which can be utilized during the peak period. A further reference to the Weather Bureau records indicates that maximum conditions prevail during the day for not more than three hours, and consequently the refrigerating system can be run for a longer period at maximum efficiency with tanks to store cold water or brine for supplementing the actual output of the refrigerating equipment when the load is more than the machine will carry. This situation brings complications. Storage tanks require space and extra apparatus, which increase the cost of the entire system, and further, it is difficult to determine what the size of the compressor should be because of the other variables which enter the problem. Depending upon the availability of storage space, the compressor could be designed for any reasonable percentage of the maximum load, so the smaller the compressor, the larger the storage space, and vice versa.

A third method is to provide in the compressor itself some means of reducing the capacity. This can be done by varying the speed and consequently the displacement of the compressor, or by varying the displacement, either by a partial by-pass of the cylinder or by a clearance pocket in the head of the cylinder when reciprocating compressors are used. It might be assumed that the efficiency would remain practically constant. This is not correct, inasmuch as the machine friction remains constant with the by-pass or clearance pocket method and this raises the power required per ton of refrigeration developed. Also, the volumetric efficiency of the machine falls off rather rapidly when the clearance pocket or partial by-pass is used. By varying the speed of the compressor, the efficiency of the power unit falls off as the speed is reduced, while the compressor friction remains constant. Of the two methods, the clearance pocket or partial by-pass of the cylinder is probably the more efficient for general use.

Another method of operation is the automatic starting and stopping of the refrigerating machine, with the automatic control designed to function as the load varies. This, however, is not considered good practice as mechanical troubles develop and the life of the system is impaired. If the equipment is kept in good condition, however, the machine will operate at maximum efficiency so long as it runs. The frequent starting and stopping of large compressors is liable to cause the power factor to decrease if adequate allowance is not made.

All of the methods described are used from time to time.

The methods of varying the output of a refrigeration system which have been outlined apply to the reciprocating type of compressor, although variations in the speed of the compressor to change the refrigerating output are common to all types of mechanical refrigeration.

There is a further method of controlling the compressor output which is particularly adaptable to the centrifugal type of machine. This is accomplished by varying the amount of condensing water used with the fluctuation in demand load. Because of the characteristics of the centrifugal

type of apparatus, as the condensing water quantity is reduced and the condensing temperature consequently raised, the discharge pressure of the centrifugal machine rises correspondingly and the horsepower input to the machine falls off. While this reduces the total power input to the machine, it does not necessarily reduce the power input per ton of refrigeration developed, as the power input does not drop with a rising discharge pressure as fast as the refrigerating effect produced drops. It is a method, however, which shows marked economies over the method generally used by the operating engineer, which is to lower the suction pressure in order to reduce the refrigerating output of the system.

Steam Jet System

So far the discussion has been confined to reciprocating, centrifugal, and rotary compressors. The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. Fig. 3 shows a complete flow diagram of the system. The power used for compressing

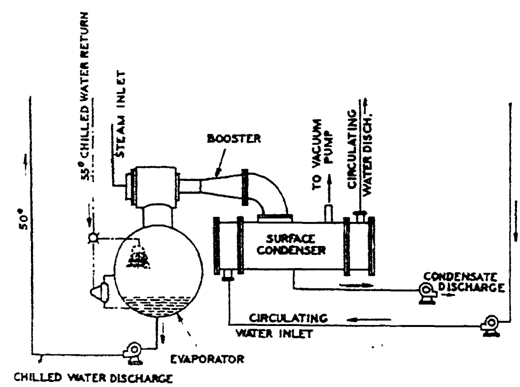

FIG. 3. DIAGRAM OF STEAM JET REFRIGERATION UNIT

the refrigerant is steam, taken directly from the boiler, thus eliminating the mechanical losses of manufacturing electric current. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiencies of the equipment are somewhat lower than those of the positive mechanical type of compressor; also the condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures, and steam ejectors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

Fig. 3 shows a typical water cooling application. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam, and as this requires heat and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a

temperature corresponding to the boiling point, determined by the vacuum maintained. The amount of water flashed into steam is a small percentage of the total water circulated through the evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired low temperature is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed into the evaporator, plus any entrained air taken out of the water circulated, to a somewhat higher absolute pressure, and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture of entrained air, evaporated water, and impelling steam is discharged into a surface condenser at a pressure which permits the available condensing medium to condense it. The resulting condensate is removed from the condenser by a small pump, from which it can be discharged to the sewer or returned to the system in the form of make-up water, or part of it may be returned to the boiler feed pump.

As the normal temperature of water required for air conditioning purposes is between 40 F and 50 F, with an average temperature of approximately 45 F, this type of water cooling is particularly desirable, as the efficiencies and operating costs compare very favorably with other types of refrigerating equipment, especially in view of the fact that the cooling apparatus is, as a general rule, less expensive to install.

Approximately three times as much condenser water is required for the steam jet cooling system as would be necessary with other types of mechanical refrigeration, but as the system can be designed with a large number of jets, each of which can be cut off as the load falls below maximum, constant refrigerating efficiency is maintained and frictional losses and volumetric inefficiencies are kept at a minimum.

The slight amount of air which may be entrained in the cooled water is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A small secondary condenser, of course, is necessary to condense the steam used in the secondary jet.

Steam jet refrigeration has an advantage where cooling towers are used for supplying the condensing liquid, as there is a great saving in the amount of steam used per ton of refrigeration. As the outdoor weather conditions vary the load on the cooling system, the compression ratio between the condenser and evaporator can be reduced and less propelling steam need be used per ton of refrigeration developed. Roughly, in air conditioning work, mechanical compressors show a falling off of 30 to 40 per cent in the power input when using the most economical arrangement of compressors, as the load varies from 100 per cent to 25 per cent of the rated capacity; whereas with steam jet cooling equipment, the amount of steam required for producing the necessary refrigerating effect falls off in direct proportion to the load on the system. When steam refrigeration is employed with cooling towers, the efficiency increases as the output is reduced.

Compressors and Refrigerants

There are many different types of compressors, a number of refrigerants, different types of evaporators, condensers and arrangements of the cycle, and each type has its particular place and usage.

The generally used compressors are of the following types:

1. Reciprocating compressors.
2. Centrifugal compressors.
3. Rotary compressors.
4. Steam jet compressors.

Over-all efficiency of the compressor in smaller commercial installations is not as important a requirement as that the whole unit require little attention and make a minimum of noise. The noise level when the fan, sprays, and compressor are in full operation should not exceed 25 decibels. High compressor efficiency appears as an important factor only in the larger industrial air conditioning systems.

The refrigerants in most general use in commercial and industrial air conditioning are here listed in the order of their inoffensive odor characteristics:

1. Water vapor.
2. Carbon dioxide.
3. Dichlorodifluoromethane.
4. Dichloromethane, sometimes called methylene chloride.
5. Methyl chloride.
6. Ammonia.
7. Sulphur dioxide.

The various types of compressors bear varied relationships to the refrigerants used in both commercial and industrial air conditioning. *Reciprocating compressors* are generally used for any of the refrigerants listed except water vapor, dichloromethane, or other low pressure refrigerant, and they are used in both commercial and domestic air conditioning systems. They have been developed to a point where their efficiency is high and their operation very satisfactory. Relatively low speed operation makes them desirable for general use in large installations. They are of two types, vertical and horizontal, either single or double acting. The horizontal double-acting compressor is not generally used in air conditioning except when carbon dioxide is used as the refrigerant in the larger industrial systems. Vertical, single-acting, encased crank, reciprocating compressors of the uniflow type with valves in the pistons have proven reliable and are used in capacities from 1 hp to more than 100 hp. Reciprocating compressors can be used with more refrigerants than other types of compression units. For instance, when carbon dioxide is used as the refrigerant, a reciprocating compressor is required because of the extremely high pressures and the relatively high ratio of compression.

The production of refrigeration at temperature levels from 25 F to 55 F for general air conditioning involves special types of refrigerating compressors. Among these are:

1. Centrifugal compressors using a volatile refrigerant.
2. Centrifugal compressors using water as a refrigerant.
3. Steam jet or vacuum systems using water as a refrigerant.
4. Rotary compressors using a volatile refrigerant.

The first two types, *centrifugal compressors*, using dichloromethane or water vapor, can theoretically be used with any of the other refrigerants,

but the resulting loss in efficiency with the higher pressure gases limits the centrifugal compressor to the two refrigerants named. At the present time the centrifugal compressors are limited to air conditioning systems of about 75 hp and more. Centrifugal compressors are usually built in two or more stages where the compression ratio is high, and their design follows closely that of any other centrifugal equipment, such as general service pumps and fans.

Steam jet compressors which have recently entered the field are simple and compact and, having no moving parts, they produce practically no vibration but are not economical for water temperatures much below 40 F or where the cost of generating steam is higher than the cost of operation with other prime movers.

Rotary compressors are generally used for methyl chloride and dichlorodifluoromethane because of their relatively low pressure and compression ratios. These compressors find widest use for fractional tonnage duty.

The source of condensing water to some extent governs the type of refrigerant used. If condensing water is available at temperatures of not more than 70 to 75 F any of the refrigerants mentioned can be used economically, but if the available condensing water temperature is above 80 F, carbon dioxide becomes uneconomical as its critical temperature is approximately 88 F. A condensing water temperature over 80 F makes the power required for compression high. All refrigerants have critical temperatures and pressures sufficiently high so that their efficiency is not materially affected by the condensing water temperatures, except in so far as this temperature affects the compression ratio. Steam jet cooling systems can use water up to 85 F, or even slightly warmer.

The applicability of the various refrigerants is interesting. Carbon dioxide is limited by the condensing water temperature; the power consumption is slightly higher than that of other refrigerants; and the pressures are three to four times that of ammonia.

The condenser pressures of methyl chloride and dichlorodifluromethane are approximately one-half that of ammonia.

Ammonia, probably the best known refrigerant, has the disadvantage of being toxic, and under certain circumstances explosive, corrosive, and irritating, even in small quantities in the atmosphere. Ammonia is used exclusively in the larger indirect or brine cooling air conditioning systems.

Sulphur dioxide is corrosive and irritating even in small quantities in the atmosphere and it is toxic under certain circumstances.

Dichloromethane operates at pressures below that of the atmosphere, and it is to some extent toxic.

Dichlorodifluromethane under normal circumstances is non-toxic, non-irritating, and non-explosive, but under high temperatures it breaks down into several obnoxious, poisonous components.

Methyl chloride, under certain conditions, is explosive and slightly toxic.

The steam ejector water vapor system has none of the disadvantages of toxicity, explosiveness and corrosiveness encountered in the other refrigerants, but the system operates at less than atmospheric pressure. This, however, is not an important factor as there are no moving parts in the compressor and the possibility of inleakage of air is remote as all of the

equipment can be welded air and water tight. The supply of water is inexhaustible, and as a refrigerant, the make-up cost is negligible. The same boiler equipment can be used for heating in winter and for cooling in summer.

Electric Motors

The motors used for driving compressors can be roughly classified in three groups: synchronous, multispeed, or variable speed. Further information on motors may be found in Chapter 17.

Coolers

The types of coolers used in connection with air conditioning work fall into three general groups. The *first* is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. One method of the direct cooling of water is to install direct expansion coils in the spray chamber so that the water sprayed into the air comes in direct contact with the cooling coils. Another common and efficient method of cooling spray water is to use a Baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant.

Another type of spray water cooler is the shell and tube heat exchanger in which the refrigerant is expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell completely surrounding the tubes at all times, good contact and a high rate of heat transfer are insured. The disadvantage of such a system is that with the falling off of load on the compressor the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, might split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than for any other type of cooler. However, with large storage tanks this type of cooling can be utilized to advantage.

When direct cooling of air is employed, the refrigerant is inside the coil and the air passes over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used can be either smooth or finned, the finned coil being more economical in space requirement than the smooth coil. The fins, however, must be far enough apart so as not to retain the moisture which condenses out of the air.

The indirect cooler, where brine is cooled by the refrigerant and the resulting cold brine is used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and that of the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of

refrigeration increases due to the higher compression ratio, but there are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used, due to fire or other risks especially in densely populated areas, the brine can be cooled in an isolated room or building and then be circulated through the air conditioning equipment in perfect safety because it is used to cool the water or air, without any possibility of direct contact between the air and refrigerant.

When an indirect system of cooling is used, it will be found that the heat transfer rate of the water cooler is considerably higher as a general rule than that of a direct expansion cooler for the same requirements. With direct expansion interchangers, it is almost impossible to keep the entire system flooded with liquid, whereas with brine interchangers the cooling medium completely fills the space of the interchanger and perfect contact is insured.

Ice may be used for chilling water or air for conditioning work. Its application is limited because of the cost of ice, although the efficiency of cooling is higher than any other water cooling system. The word "water cooling" is used advisedly in that the direct cooling of air by ice is, while not impossible, rather impractical. It might be said that ice coolers are economical for systems requiring a maximum of 20 tons per 24 hours where the load fluctuates considerably, and it is possible to introduce ice only as it is required to cool water. The most general method of cooling water with ice is to spray the water over the surface of the ice, insuring as much contact as possible and approximating the same performance as the Baudelot type of cooler. Because of the large fluctuations in load in the air conditioning system, the higher cost of refrigerating effect when ice is used is offset by the fact that there are no motor and condenser inefficiencies under partial load. Also, because the cost of the mechanical refrigeration equipment for the small system is so much higher per unit of effect, the fixed charges are small enough to overbalance the extra cost of the ice.

Condensers

Condensers are usually either the double pipe type or the shell and tube type. Shell and tube condensers are almost identical with coolers. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes, and refrigeration passes through the annular space in the outer pipe. Where possible, there should be counter flow of the refrigerant and the condensing water to maintain maximum temperature differences.

The amount and temperature of the condensing water determine the condensing temperature and pressure, and indirectly the power required for compression. It is, therefore, necessary to strike a balance so that the quantity of water insures economical compressor operation.

As part of the condenser, or attached to it, there must be storage space for liquid refrigerant. The installation of all equipment should be made accessible for inspection, repair, and cleaning. Both the coolers and condensers should have space for pulling tubes.

Because there is a decided tendency to conserve the water in city mains and most large cities are restricting the use of water, in order to use air

conditioning systems and refrigeration equipment it is often necessary to install cooling towers. The cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet not only the maximum load at normal conditions, but also the maximum load at abnormal condensing water temperatures. If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains in that the temperature of the condensing water varies directly with the outdoor temperature and, as pointed out, the refrigeration load also varies with this temperature. Certain economies are possible when a cooling tower is used which cannot be achieved by the use of condensing water from city mains, even where the city water temperature is extremely low. Normally, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of what the outdoor temperatures are. With the cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water off the cooling tower drops considerably, and it has been established that 50 per cent of the time the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, therefore, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is negligible, as the only water required is that used to make up the loss by evaporation in the cooling tower itself. See also Chapter 11.

PROBLEMS IN PRACTICE

1 ● In a locality where the electric power rate is based on a demand charge, it is desired to install the smallest possible compressor motor which will provide summer cooling for a 300-seat restaurant which operates 6 hours per day from 11 a.m. to 2 p.m., and from 5 p.m. to 8 p.m. The refrigeration load at the peak is 28 tons. If the load factor for both the noon and evening meals is 70 per cent, discuss the type of equipment which would take the greatest advantage of the reduced power rate at low kilowatt demand.

A storage system using a chilled water storage tank would permit the installation of a refrigeration system having the smallest motor.

For a 28-ton system operating 6 hours per day at a 70 per cent load factor, on the maximum day the total heat removed would be,

$$28 \text{ tons} \times 6 \text{ hr} \times 0.7 = 117.5 \text{ ton-hours per day.}$$

If a compressor were to operate 24 hours at a constant rate, its average capacity would be $\frac{117.5 \text{ ton-hours}}{24 \text{ hours}} = 4.9$ tons, or approximately 5 tons. If operated 12 hours per day, the compressor capacity would have to be increased to 10 tons.

A water storage tank would store the refrigeration and allow off-peak operation, so a smaller compressor motor could be used. However, the suction temperature at which the compressor would be operated would be lowered approximately 5 to 10 F. This would increase the horsepower per ton of refrigeration, when dichlorodiflouromethane is used, approximately 10 per cent for a 5 F reduction and 24 per cent for a 10 F reduction in the

suction temperature. Rather than store the water at too cold a temperature, it would be more economical to install a larger storage tank and use a higher temperature.

A 5-ton compressor running during periods when there are no customers, namely, during the 15 hours from 8 p.m. to 11 a.m. will have stored 15 hr × 5 tons, or 75 ton-hours, of refrigeration in the storage tank by 11 a.m. As one ton-hour equals 12,000 Btu, 75 × 12,000 Btu, or 900,000 Btu, will have been stored.

If the apparatus dew-point temperature is 54 F, and the chilled water is supplied to the air washer at 48 F, it will leave at 54 F. If the water in the storage tank is at 40 F, the temperature difference between the stored water and the water entering the washer will be 48 F − 40 F = 8 F. This is equivalent to an available 8 Btu of cooling effect per lb of water stored. Therefore, $\frac{900{,}000}{8}$ or 112,500 lb of water must be stored. This is $\frac{112{,}500}{8.34}$ = 13,500 gal water to be stored, which equals $\frac{13{,}500}{7.5}$ = 1800 cu ft of water.

The storage tank to hold this water might be 6 ft high, 7½ ft wide, and 40 ft long. Should this volume prove impractical, a proportionately smaller tank could be used if the water storage temperature were reduced. Should a 10-ton refrigeration system be used, the water quantities and tank capacity could be reduced by one half, and the refrigeration plant need not be started until 8 a.m. daily, which might prove of additional advantage.

If refrigeration is stored by freezing ice on coils, considerable storage space will be saved but more power input per Btu of cooling will be required.

2 ● For condensing purposes, an air conditioning system uses city water which has an average 70 F supply temperature. The following table lists the number of hours per year during which definite wet-bulb temperatures and corresponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 − 75	100	233
74 − 70	277	183
69 − 65	330	157
64 − 60	277	144
59 − 55	158	79
54 − 50	52	37
	Total 1200 hours	

If the power requirements of a dichlorodifluoromethane refrigeration system are in accordance with the following data on partial load operation, determine the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Kw per ton	**0.89**	**0.89**	**0.87**	**0.86**	**0.86**	**0.93**	**0.97**

Seasonal power cost:

Wet-Bulb Temperature F	Ton-Hours	Kwhr
80	6 × 284 = 1,704	1,704 × 0.89 = 1,517
79 − 75	100 × 233 = 23,300	23,300 × 0.89 = 20,750
74 − 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 − 65	330 × 157 = 51,800	51,800 × 0.86 = 44,500
64 − 60	277 × 144 = 39,900	39,900 × 0.86 = 34,300
59 − 55	158 × 79 = 12,500	12,500 × 0.93 = 11,600
54 − 50	52 × 37 = 1,920	1,920 × 0.97 = 1,860
Totals	181,824 ton-hours	158,627 kwhr

The 158,627 kwhr at 2 cents per kwhr will cost \$3,173.

The average consumption will be $\frac{158{,}627 \text{ kwhr}}{181{,}824 \text{ ton-hours}} = 0.873$ kw per ton.

3 ● **Using the data from Question 2, if city water costs 20 cents per thousand gallons, and if 1.25 gallons are used per minute per ton, estimate the annual water cost.**

$60 \times 1.25 = 75$ gal per ton-hour.

$181{,}824$ ton-hours $\times 75 = 13{,}620{,}000$ gal per year.

$\frac{13{,}620{,}000 \times \$0.20}{1000} = \$2{,}724$, the yearly cooling water cost.

4 ● **Using the data of Question 2, if a cooling tower were installed for re-using the condensing water, estimate the annual operating cost of a dichlorodifluoromethane refrigeration system if the final temperatures of the water leaving the cooling tower and the kilowatt input per ton are the following:**

Tons	**284**	**233**	**183**	**157**	**144**	**79**	**37**
Temperature of water leaving tower, F	**86.7**	**81.8**	**76.5**	**72.1**	**66.4**	**61.3**	**55.6**
Kw input per ton	**1.10**	**0.94**	**0.85**	**0.80**	**0.74**	**0.59**	**0.62**

Wet-Bulb Temperature F	Ton-Hours		Kw per Ton		Kwhr
80	1,704	×	1.10	=	1,875
79 – 75	23,300	×	0.94	=	21,900
74 – 70	50,700	×	0.85	=	43,300
69 – 65	51,800	×	0.80	=	41,400
64 – 60	39,900	×	0.74	=	29,500
59 – 55	12,500	×	0.59	=	7,370
54 – 50	1,920	×	0.62	=	1,200
Totals	181,824 ton-hours				146,545 kwhr

The 146,545 kwhr at 2 cents per kwhr will cost \$2,931.

The average consumption will be $\frac{146{,}545 \text{ kwhr}}{181{,}824 \text{ ton hours}} = 0.805$ kw per ton.

5 ● **If a steam ejector system were used to secure the refrigeration for the air conditioning system of Question 2, compute the annual steam cost if steam is sold for 53 cents per thousand pounds and if there is an average steam consumption of 20 lb of steam per hour per ton when used with a cooling tower system.**

181,824 tons $\times$ 20 lb of steam per ton $= 3{,}636{,}480$ lb of steam.
The 3,636,480 lb at 53 cents per thousand pounds will cost \$1,929.

6 ● **From the data given in the following table covering auxiliary equipment, make a comparison between the operating costs of the complete dichlorodifluoromethane system of Question 4 and the complete steam ejector cooling system of Question 5. A cooling tower is used for condenser water recovery.**

Plant Operation	Dichlorodifluoromethane System	Steam Ejector System
Hours of operation	**1200**	**1200**
Cooling tower fan, bhp	**17.8**	**35.6**
Cooling tower pump, bhp	**30.2**	**47.8**
Chilled water, gpm	**1200**	**1200**
Discharge head on chilled water system, ft	**75**	**75**
Pump efficiency, per cent	**75**	**75**
Motor efficiency, per cent	**80**	**80**
Chilled water temperature, F	**46**	**46**

The flash tank or evaporator of the steam ejector system is of the open type, the flash water being pumped directly to the sprays of the washer used for cooling the air.

Dichlorodifluoromethane System:

Power requirements,

Cooling tower fan	17.8 bhp
Cooling tower pump	30.2
Total	48.0 bhp

$$\text{Power for cooling tower system} = \frac{48.0 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 53{,}700 \text{ kwhr.}$$

The water cooler in a dichlorodifluoromethane system of the surface type requires no additional pumping head other than the friction drop through the cooler, which in this problem is estimated to be 10 ft. The total pumping head is, therefore, 75 + 10 = 85 ft. Power required for the chilled water system will be,

$$\frac{1200 \text{ gpm} \times 8.34 \text{ lb per gallon} \times 85 \text{ ft head}}{33{,}000 \text{ ft lb} \times 0.75 \text{ pump efficiency}} = 34.3 \text{ bhp.}$$

$$\frac{34.3 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 38{,}300 \text{ kwhr.}$$

Thus, the total power required by the auxiliary equipment will be

$$53{,}700 + 38{,}300 = 92{,}000 \text{ kwhr.}$$

The 92,000 kwhr at 2 cents per kwhr will cost	$1,840
The power cost of refrigeration, from Question 4, is	2,931
The total annual power cost, using a dichlorodifluoromethane system, is	$4,771

Steam Ejector System:

Power requirements,

Cooling tower fan	35.6 bhp
Cooling tower pump	47.8
Total	83.4 bhp

$$\text{Power for cooling tower systems} = \frac{83.4 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 93{,}300 \text{ kwhr.}$$

In the flash tank or water cooler of the steam ejector system, the water is at a pressure corresponding to the chilled water temperature required. In this case it is at 46 F, which corresponds to an absolute pressure of 0.1532 lb per sq in. or 0.3118 in. Hg. This increases the pumping head on the chilled water circulating pump by 14.7 − 0.15 = 14.55 lb per square inch, or 33.5 ft. The total pumping head is, therefore, 75.0 + 33.5 = 108.5 ft.

$$\frac{1200 \text{ gpm} \times 8.34 \text{ lb per gallon} \times 108.5 \text{ ft head}}{33{,}000 \text{ ft-lb} \times 0.75 \text{ pump efficiency}} = 43.7 \text{ bhp.}$$

$$\frac{43.7 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 48{,}800 \text{ kwhr.}$$

The total power required by the auxiliary equipment is

$$93{,}300 + 48{,}800 = 142{,}100 \text{ kwhr.}$$

The 142,100 kwhr at 2 cents per kwhr will cost	$2,842
The cost of the steam, from Question 5, is	1,929
The total annual power cost, using a steam ejector system, is	$4,771

These calculations indicate that for the assumptions made, both the dichlorodifluoromethane system and the steam ejector system would cost 2.6 cents per ton-hour to operate. In order to obtain a complete analysis it would be necessary to compare the fixed charges which include interest, depreciation, obsolescence, and maintenance. These are customarily computed at 15 per cent of the initial cost per annum. To this cost must be added the cost of refrigerant make-up per year. In the steam system this is negligible, but in the dichlorodifluoromethane system it may be approximated at from ¼ to ⅓ of the refrigerant charge per year.

Chapter 11

HUMIDIFICATION AND DEHUMIDIFICATION

Air Washers, Atmospheric Water Cooling Equipment, Cooling Towers, Design Wet-Bulb Temperature, Cooling Ponds, Natural Draft Deck Type Towers, Mechanical Draft Towers, Winter Freezing

EQUIPMENT for humidifying and dehumidifying is of varied character and its functions will be discussed in this chapter. An air washer is essentially a chamber in which air is brought in intimate contact with water, the object being (*a*) to wash the air or (*b*) to regulate the moisture content of the air and at the same time wash it. The air comes in contact with the water by passing it through water sprays or by passing it over surfaces wetted by a continuous flow of water; hence the classification: spray, scrubber, and combination spray and scrubber type washers.

A washer chamber may be constructed of wood, or stone, but it is most often constructed of sheet metal. The lower portion of it is specially designed as a tank to receive the water dropping through the chamber and to serve as a reservoir from which the water may be recirculated.

It is desirable that air leaving a washer contain no water in suspension. For this reason eliminators are provided at the washer outlet. These may be in the form of plates or baffles upon which the free moisture is deposited as the air is deflected through several changes from its original direction of flow. In some washer units steel wool filter sections serve as eliminators. However, specially designed plates are used more generally than other devices because they offer the least resistance to the flow of air, while still performing effectively the function of free moisture elimination. They also have the advantage of acting as scrubber surfaces when flooded.

It is essential to uniform performance in a washer, that air enter evenly distributed over the washer inlet. To insure this, a perforated plate or eliminator plates are installed at the inlet. Eliminator plates are now more generally used. They serve a second purpose in preventing the escape of spray through the washer inlet.

Water is supplied to scrubber type units through flooding nozzles. The capacity of these nozzles varies with the manufacturer although a fair value of 5 gpm may be used. The nozzles are spaced on one-foot centers across the top of the washer over the scrubber plates.

Water is supplied to spray type units through atomizing nozzles generally arranged in banks across the washer. The nozzles spray either in the direction of the air flow, that is, downstream, or against the air flow, or upstream. Nozzle capacities vary with the manufacturer, from 1-½ to 2 gpm at a water pressure of about 25 lb per square inch which pressure

is required for effective atomization. The spacing of spray nozzles is determined by the water requirements of the particular installation. A spray type washer may contain one, two or three banks of nozzles depending upon its application.

When an air washer is used for cleaning air it removes impurities and dusts. In general it does not function as efficiently in this service as a filter. For non-microscopic soluble dust its efficiency averages about 50 per cent, unless the concentration of dust is high. Its effectiveness in removing greasy microscopic dust is practically negligible as is also its deodorizing ability.

When a washer is used to regulate the moisture content of air it adds moisture to (humidifies) or removes moisture from (dehumidifies) the air to achieve the desired moisture content. (See also Chapter 3.)

When air passes through a washer wherein water is circulated without the addition or removal of heat, the air tends to become saturated at its entering wet-bulb temperature. What occurs here is partial or complete adiabatic saturation. The total heat content of the air is unchanged, inasmuch as the dry-bulb temperature of the air drops in proportion to the amount of additional water evaporated. This action is also known as evaporative cooling. A measure of the washer's effectiveness under these conditions is its saturating efficiency which is equal to the drop in dry-bulb temperature in per cent of the entering wet-bulb depression. Other things being equal, the saturating efficiency of a spray type washer is a function of the number of spray banks and the direction in which they spray. The following table gives a general comparison:

3 banks—2 upstream—1 downstream	100% saturation efficiency
2 banks—2 upstream	95% saturation efficiency
2 banks—1 upstream—1 downstream	85% saturation efficiency
1 bank —upstream	80% saturation efficiency
1 bank —downstream	65% saturation efficiency

When air passes through a washer wherein the circulated water is either cooled or heated before being returned to the spray chamber, a heat interchange between the air and water occurs, and the air tends to become saturated at the temperature of the leaving water. The extent to which the leaving air and leaving water temperatures approach each other is an index to the effectiveness of the washer under the operating conditions. The total heat absorbed by the water in the process equals the total heat given up by the air, or the heat given up by the water equals the heat absorbed by the air. Depending on whether the moisture content of the air is increased or decreased during the operation, humidification or dehumidification occurs. Heat will be added to or removed from the air as the water supplied is of a higher or a lower temperature than the wet-bulb temperature of the entering air.

For dehumidifiers the ratio of the difference between the leaving wet-bulb and the leaving water to the difference between the entering wet-bulb and the entering water may be figured as follows:

3 banks—1 downstream—2 upstream	0%
2 banks—2 upstream	5%
2 banks—1 upstream—1 downstream	15%
1 bank —upstream	20%
1 bank —downstream	35%

Humidifiers may be figured on the same basis as dehumidifiers; the leaving water temperature, of course, will be higher than the wet-bulb temperature of the leaving air.

The problem of cooling or heating the circulated water before returning it to the washer chamber is external to the unit. It will suffice here to note that heating is generally accomplished by passing the water through closed hot water heaters or by injecting steam into the water circuit; cooling, by passing the water through closed coolers or over refrigerating coils in a Baudelot chamber. Often in a cooling and dehumidifiying application, the refrigerating coils are located within the washer chamber.

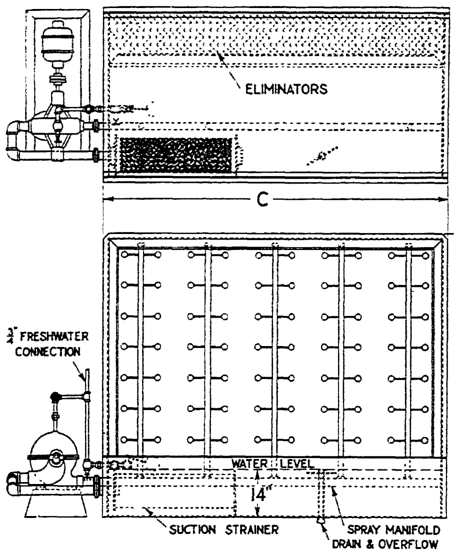

FIG. 1. TYPICAL SINGLE BANK AIR WASHER

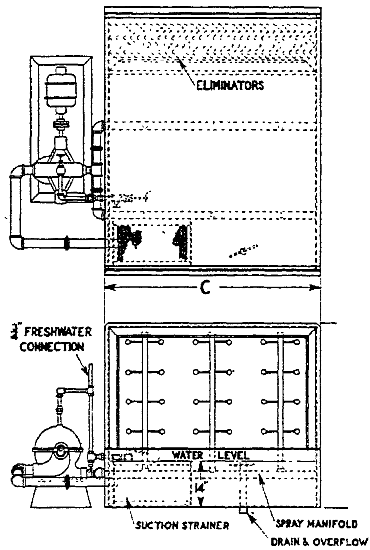

FIG. 2. TYPICAL TWO BANK AIR WASHER

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the over-all efficiency of heat transfer between air and the cooling or heating medium (water, brine, etc.). A multi-stage washer is equivalent to a number of washers in series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 feet per minute through the gross cross-sectional area of the unit above the water level in its tank. At this rating spray type washers handle about 2-½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed. Two of the more important items are:

a. Choose a washer for air velocities above approximately 300 fpm and below approximately 600 fpm. Velocities outside this range are likely to result in faulty elimination of entrained moisture.

b. When a high saturating efficiency is required, select a two or three bank spray type unit, having a total water capacity of not less than 15 gpm per 100 cfm.

The area of a washer may be dictated by space limitations outside the washer, such as headroom, or by space requirements inside washer, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays, (the *leaving* eliminators require about 1 ft 6 in., *entering* eliminators about 1 ft).

The resistance to air flow through an air washer varies with the type eliminators, number of banks of sprays, direction of spray, type of scrub-

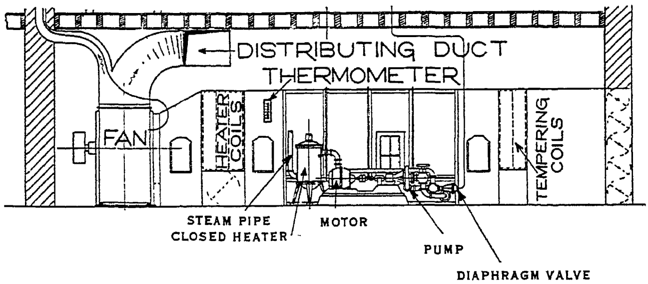

FIG. 3. AIR WASHER WITH SPRAY WATER HEATING ARRANGEMENT

ber plates, and, if cooling coils are located in unit, by their size and type. Washers should be selected to limit static resistances below 0.50 in.

Power Requirements

The approximate power requirement for passing 10,000 cfm of air through a humidifier of the spray type by a fan of 78 per cent mechanical efficiency is given in Table 1, this being the fan brake horsepower for various velocities and static pressure losses. Allowance should be made for variations in static pressure due to the use of different diffuser plates or inlet louvers and for variations in fan efficiencies.

ATMOSPHERIC WATER COOLING EQUIPMENT

To successfully operate a refrigerating plant or a condensing turbine, the heat from the compressed refrigerant or the discharged steam must be removed and dissipated. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger. If the plant is situated on the banks of a river or lake, an intake may be had upstream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of water is a city supply or well water, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains

dissolved salts which would quickly form scales on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

Table 1. Approximate Fan Brake Horsepower

Requirements for passing 10,000 cfm of air through humidifiers at various velocities and static pressures. Mechanical efficiency of fan—78 per cent.

Velocity FPM	30 Deg Eliminators Spaced on 1-1/8 In. Centers		45 Deg Eliminators Spaced on 2-1/4 In. Centers	
	Static Pressure In. Water	BHP	Static Pressure In. Water	BHP
500	0.20	0.40	0.40	0.80
550	0.24	0.48	0.48	0.97
600	0.29	0.58	0.58	1.15
650	0.34	0.68	0.68	1.35

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1.) Because it is impractical to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

Cooling Towers

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess quantity of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

TABLE 2. CONDENSER DESIGN DATA

GAS	MAXIMUM PRESSURE DESIRED IN CONDENSER	GAS TEMPERATURE IN CONDENSER F	LEAVING HOT WATER TEMPERATURE F	
			Best Design	Average Design
Steam	28 in. vacuum	99.7	97	93
Steam	27 in. vacuum	114.3	110	105
Steam	26 in. vacuum	126.0	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodi-fluoromethane	117 lb gage head pressure	100.0	96	93

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 deg or 4 deg of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 deg difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.
2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1931, inclusive, there were but 6 hrs per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September, inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City), with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 8, shows safe design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}.$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 3 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various parts of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton
Condenser turbine	950 to 980 Btu per pound of steam
Steam jet refrigerating appartus	1030 to 1150 Btu per pound of steam
Diesel engine	2800 to 4500 Btu per horsepower

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line, as natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

EQUIPMENT	COOLING EFFICIENCY—PER CENT		
	Minimum	Usual	Maximum
Spray Ponds	30	45 to 55	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift off. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased

volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles, to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 ft to 6 ft above the edge of the basin, to supply from 5 lb to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground if they have an earthen or a concrete basin, or they may be placed on roofs having special waterproof roofing. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae formations may be a considerable nuisance in a spray pond. Such growths are killed by the periodic addition of potassium permanganate to the pond water. Addition of the dissolved chemical should be made until the water holds a faint pink color for at least 15 min.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The

louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 ft to 80 ft high and from 6 ft to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1 Chapter 8, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case should it need a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water

just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of louvered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 fpm to 600 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside. Such apparatus does not have the counter flow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

Make-up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 deg range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to avoid excessive deposits in the condensers.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and

collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck

TABLE 4. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	COOLING POND	SPRAY POND	SPRAY TOWER	DECK TOWER	MECHANICAL DRAFT	INDOOR TOWER
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per sq ft	x	x	1	3	4	2
Independence of wind velocity	6	3	4	5	1–2	1–2
Drift nuisance	1	6	5	4	2–3	2–3
Make-up water required	1	6	5	4	2–3	2–3
Pumping head	1	2	3	4–5	4–5	6
Maintenance	2	1	3	4	5	6
Suitability for congested districts	x	5	4	3	1	2
Water quantity required for definite result	6	5	4	1–2	1–2	3

xNot comparable.

tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 4.

PROBLEMS IN PRACTICE

1 ● What three systems of humidification are used in textile, printing, and lithographic plants?

a. Indirect: Introduction of moistened air into the rooms.

b. Direct: Spraying of moisture into the rooms.

c. Combined: Direct and indirect as above.

2 ● How may relative humidity be controlled?

a. If constant room temperature is to be maintained:

1. To maintain a constant relative humidity, the dew point must be kept constant.

2. To increase the relative humidity, the dew point must be raised.
3. To decrease the relative humidity, the dew point must be lowered.

b. If constant dew point is to be maintained:
1. To maintain a constant relative humidity, the room temperature must remain constant.
2. To increase the relative humidity, the room temperature must be lowered.
3. To decrease the relative humidity, the room temperature must be raised.

c. With varying dew-point temperatures:
1. To maintain a constant relative humidity, the room temperature must vary directly and in almost equal amount with the dew point.
2. To increase the relative humidity, the difference between room temperature and dew point must be decreased.
3. To decrease the relative humidity, the difference between room temperature and dew point must be increased.

d. With varying room temperatures:
1. To maintain a constant relative humidity, the dew point must vary directly and in almost equal amount with the room temperature.
2. To increase the relative humidity, the difference between dew point and room temperature must be decreased.
3. To decrease the relative humidity, the difference between dew point and room temperature must be increased.

3 ● In industrial air conditioning plants, what are the four sources of heat which must be taken into consideration in the design of a system?

a. Heat transfer from the outside air.
b. Body heat from employees.
c. Sun effect.
d. Heat equivalent of power consumed in driving machinery, in lighting, and in manufacturing processes in general.

4 ● Why do cooling towers give best results when the humidity of the air is low?

The cooling of water by dropping it through air depends mostly upon the evaporation of the water. If the relative humidity of the air is low, the water vapor will be readily absorbed and carried away, while if the humidity of the air is high, its capacity to pick up water vapor is less and the water is cooled less with the same exposure to air.

5 ● What performance tests should be given air washers?

a. Capacity.
b. Resistance.
c. Visible entrainment of free moisture.
d. Humidifying efficiency.
e. Cleaning effect.

6 ● What are the several different types of water-cooling towers?

a. Those with forced draft.
b. Those with natural draft open to the atmosphere.
c. Those with natural draft closed to the atmosphere.
d. Those with combined natural and forced draft.

7 ● What are the different types of air washers?

a. Spray. *b*. Wet scrubber. *c*. Combination spray and scrubber.

8 ● What is the saturation efficiency for an air washer with the common variations in spray arrangement?

For three banks, two up-stream and one down-stream	100%
For two banks, both up-stream	95%
For two banks, one up-stream and one down-stream	85%
For one bank, up-stream	80%
For one bank, down-stream	65%

9 ● Upon what air velocity are air washers usually rated?

500 fpm, through the area above the tank.

10 ● What wet-bulb temperature for the outside air is usually selected in air conditioning design when cooling is to be accomplished?

One which is not exceeded more than 5 to 8 per cent of the time in the locality where the plant is to be situated.

11 ● Where should the suction connection be placed in a cooling pond?

As far below the surface as possible and as far away from the discharge as practicable

12 ● What chemical is used to kill algae formations in spray ponds?

Potassium permanganate.

13 ● What is the usual amount of spray water delivered to a cooling pond per square foot of pond area?

From 0.1 gpm on small sizes to 0.8 gpm on large sizes.

14 ● What is the usual amount of water delivered in cooling towers per squar foot of area?

From 0.6 to 1.5 gpm.

15 ● About how much water is lost by evaporation in atmospheric cooling?

About 1 gal per 1000 gal for each degree of cooling range.

16 ● How is freezing obviated in cooling pond sprays?

The pressure and quantity of water is lowered so that the drops become of increased size and do not freeze so readily.

17 ● What is the cause of a high concentration of salts in the cooling water of an atmospherically cooled system?

The constant evaporation of a small portion of the water leaves salts behind to accumulate in the unevaporated water.

Chapter 12

UNIT AIR CONDITIONERS AND CONDITIONING SYSTEMS

Definition, Advantages and Uses, Functions, Sources of Refrigeration and Heat, Types and Locations, Construction of Apparatus, Installation, Basis of Equipment Ratings, Calculation of Required Capacity, Approximate Costs

AIR conditioning systems fall into two general types known as the unit type and the central type. A unit air conditioner is an assembly of parts, such as fans, humidifiers, coils, controls, and other equipment, which form a complete unit at the point of manufacture. This usually restricts the size of the unit to a capacity below 10,000 cfm. With the unit conditioner, the performance is the responsibility of the manufacturer. This is in contradistinction to a central air conditioning system which may produce the same results but for which the various parts are purchased separately and assembled by the contractor on the job, who guarantees the performance of the assembled system.

Unit Air Conditioner

A *unit air conditioner* generally has a capacity less than 30,000 Btu per hour for cooling, or 60,000 Btu per hour for heating, to make it suitable for the space to be conditioned. If it does not provide simultaneous control of at least four of the recognized functions of air conditioning (see p. 201) the apparatus should be classified as a unit heater or unit ventilator (Chapter 13) or as a unit cooler, a humidifier, or a window-type ventilator.

The apparatus, instead of being wholly self-contained, may depend upon separately located parts piped to supply heating, cooling, or humidifying mediums to the unit. A duct may supply outdoor air for circulation, but ducts are seldom used for air discharge and recirculation.

When the term *unit conditioner* is applied to such set-ups as the combination of a filter and a fan in a housing to be used with gravity warm air furnaces, or to humidifiers and heating coils to be used with steam or hot water boilers to comprise a unified central air conditioning plant, the usage of the term is inaccurate; such devices may be designated as *accessory units*, but this leads to confusion. However, since such accessory equipment is used, a description and discussion of its several types are given in the next few paragraphs before the main topic of this chapter, unit heaters, is taken up.

Accessory Central Fan Conditioning Apparatus

This includes every kind of equipment constituting an accessory to an existing or new system for warm air heating service, and also certain forms of conditioning equipment used with hot water or steam boilers in residential service. Some of these accessories provide only a fan and an air filter, while others include humidifying and cooling functions. The performance of such equipment is influenced by the outside temperature and humidity; the conditions surrounding the house or apartment, such as construction and exposure to sun; the type of heating system to which the apparatus is attached; and the location of the device on the heating system. Many of these installations are of limited capacity and effectiveness; conservative manufacturers will be discriminating in their claims for added comfort from the use of such equipment, depending on its design and functions.

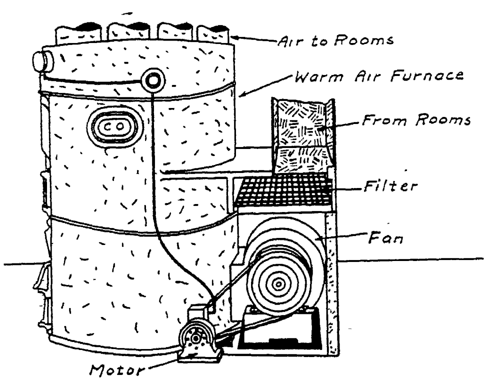

FIG. 1. FURNACE ACCESSORY UNIT

A feature of the fan-and-filter accessory unit is its availability for ventilation in summer; it makes possible a rapid cooling in the evening, after the outdoor air temperature has dropped below that of the rooms. If the fan is large enough completely to change the air in the building served every two or three minutes, the effect will be similar to that from so-called *attic fans*, (see Chapter 13), with the important advantage that the air is filtered. Fans of smaller capacity, proportioned only for the winter heating duty, may also provide an appreciable measure of cooling. Another advantage is improved headroom in the basements of residences, obtainable by substituting horizontal ducts for those of comparatively steep pitch necessary when gravity air circulation is depended upon. A fan-and-filter accessory using a dry-mat type of filter, applied to a warm air furnace, is shown in Fig. 1.

A more elaborate unit (Fig. 2), for use with a hot water heating boiler provides heating, humidification, filtering, and positive air circulation in winter; the heating coil may be used also in summer with mechanical refrigeration or for circulating city water or chilled water from an ice tank, to provide cooling and dehumidification. The disposition of fans, the cloth filter of bag design, the spray type humidifier, as well as noise

elimination features comprising canvas collars at the fan outlets and rubber pads under the fan bedplate, are indicated in Fig. 3. The use of a single element for both winter and summer functions tends to reduce the first cost, although it adds some complications in piping.

Another assembly of air conditioning equipment with a standard heating furnace, in this instance burning gas to provide warm air, is shown in Fig. 4. The apparatus comprises an air filter, a motor-driven fan, and an air washer. No refrigeration is used with this equipment.

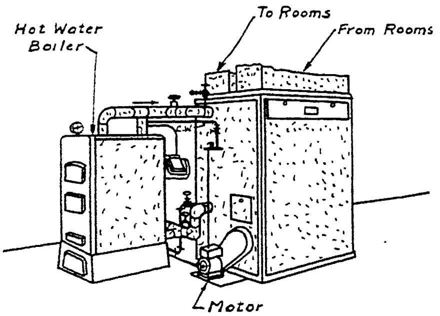

FIG. 2. UNIT WITH HOT WATER BOILER

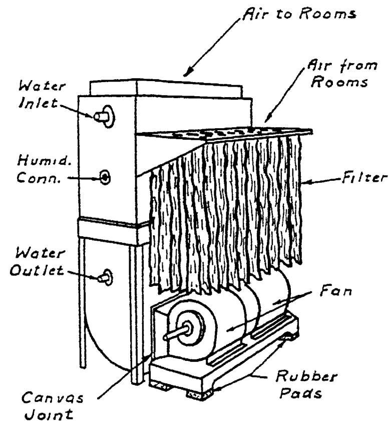

FIG. 3. HEATING AND COOLING UNIT WITH CLOTH FILTER

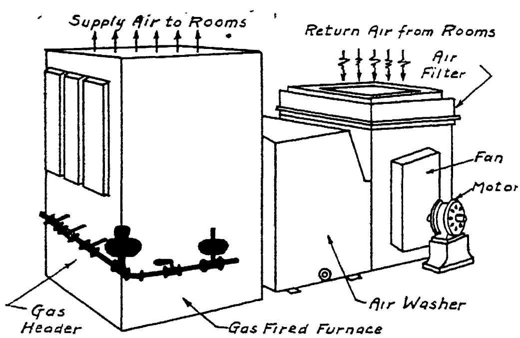

FIG. 4. GAS FIRED FURNACE UNIT

For oil fuel, the unit shown in Fig. 5 can be installed to obtain filtered, warmed, and humidified air. An oil burner and a heat exchanger provide the heat. A cooling section may be inserted between the fan and the heat exchanger, cold water being circulated through the cooling element. For automatic control, a room thermostat is provided to start the oil burner whenever the temperature falls. The rising temperature in the heat exchanger causes a second thermostat to start the fan. As soon as the temperature in the house rises to normal, the room thermostat shuts down the oil burner and operates the thermostat controlling the fan.

Having disposed of the accessory central fan conditioning apparatus, the balance of this chapter will concern only the unit air conditioner as defined in Chapter 41.

ADVANTAGES AND USES OF UNIT AIR CONDITIONERS

Unit air conditioners are suitable for commercial and comfort applications because they permit installation without seriously disturbing the building occupants, and they allow rearrangement or a change in capacity to suit changed requirements occasioned by new tenants. Tenants may even furnish their own installations and remove the apparatus from the premises at the expiration of their leases. In some types of buildings, the installation costs are lower for unit conditioners than those for central fan systems, and costs are further lowered in that there is no need for space in which to house a conditioning plant. The choice between unit and

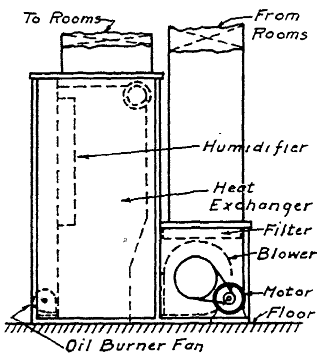

FIG. 5. OIL FIRED UNIT

central systems will, in many instances, require a close study of installation conditions at the site, and a preparation of comparative cost estimates, in addition to a consideration of the more intangible factors.

Industrial Uses

The origin of the unit conditioner, like that of air conditioning itself, was in the industrial field for maintaining desired atmospheric conditions in rooms or sections of manufacturing plants where structural limitations or service requirements made a central system uneconomic. Industrial applications continue to offer an important market for unit conditioners, in bakeries, candy factories, drug-manufacturing plants, laboratories, produce-storage rooms, printing plants, and similar places.

Commercial Uses

The most active field for unit conditioners at the present time is in commercial establishments, such as barber and beauty shops, funeral parlors, retail and specialty stores, and small restaurants, where increased patronage or larger purchases per customer offer economic justification of first cost and operating expense.

The air-handling units installed overhead in Pullman cars, diners and coaches, in the middle or at the ends of the cars which discharge the air horizontally at the ceiling level, are essentially unit conditioners. However, because of special construction to meet space limitations and other requirements, they are unsuitable for general use and are not further discussed.

Personal Uses

The major recent development in the air conditioning industry has been in new and improved types of unit conditioners suitable for apartments, homes, hotel rooms, and offices. These uses demand apparatus that is compact, of unobtrusive appearance and in harmony with the room finish and furnishings, quiet in operation, automatic, and reliable.

As with all new major appliances for the home, problems of relatively high first cost, of comparatively rapid obsolescence and of operating expense demand the continuous close attention of manufacturers and of others interested in developing the potential market. Unit conditioners are still distinctly in the pioneering stage where such problems must be met and solved if development—especially of residential units—is to proceed as fast as it should. Public understanding of residential air conditioning still requires cultivation in order to dispel fears of possible excessive operating costs and of possible high obsolescence due from frequent model changes. Progress in this direction is being helped by the increasing efforts of manufacturing companies which are now spending large sums to insure sound promotion of unit conditioners. Likewise, the *National Better-Housing Program* inaugurated in 1934 is likely to prove of real value to the air conditioning industry and to accelerate greatly the rate of public acceptance and installation of unit conditioners.

FUNCTIONS OF UNIT CONDITIONERS

Unit air conditioners may be classified as the all-year unit, the summer unit, and the winter unit. The *all-year unit* performs all of the functions of an air conditioning system; namely, cooling, dehumidification, heating, humidification, air circulation, air cleaning—with or without a supply of fresh air—and a simultaneous control of all functions. The *summer unit* must provide cooling, dehumidification, air circulation, and air cleaning; the *winter unit* must provide heating, humidification, air circulation, and air cleaning. Either of these seasonal-use units may or may not provide a fresh air supply and a simultaneous control of the functions.

In some instances, winter-type units equipped with filters for air cleaning and with fresh air connections may be operated in summer for ventilation, but the system cannot then properly be said to provide all year conditioning. It is important that the features and limitations of the specific apparatus be carefully explained to a prospective user, so that disappointments and complaints concerning operating results may be avoided.

The functions listed are performed by the unit conditioners offered by different manufacturers in various ways, some of which appear in the following outline. See the next few pages for more detailed explanations of cooling and heating theories and methods.

1. Cooling:

   By coils, usually of finned type, for direct expansion of refrigerant from a self-contained unit or from a remotely-located compressor.

   By coils, of finned type, for brine or cold water from separate refrigeration plant, or for cool water from city mains, private wells, or an ice water tank.

   By water sprays.

   By passage of air over ice cakes.

2. Dehumidification:

   By lowering the air temperature below the dew point, using any of the devices outlined for cooling.

   By adsorption materials, such as silica gel or activated alumina.

3. Heating:

   By coils, usually of finned type, for steam or hot water from system distribution mains.

   By electric heating elements.

   By gas burners.

4. Humidification:

   By evaporating or entraining water by an air current, from wetted surfaces or water sprays.

5. Air circulation:

   By motor-driven fans which discharge air into room at points, in directions and with velocities that insure adequate ventilation without drafts; air discharge usually through top, at a slight angle from vertical.

6. Air cleaning:

   By mechanical filters.

   By water washing with sprays.

   By water washing by contact with condensation or by trickling water on cooling coils or a mesh cell.

7. Fresh air supply:

   By air connection from outdoors, usually through adjustable window ducts at rear of housing, with mixing dampers for control of volume of recirculated room air taken in through louvers at each end.

8. Control:

   By manual adjustment or automatic regulation, by thermostats or hygrostats.

SOURCES OF REFRIGERATION

Mechanical Refrigeration—Direct and Indirect

In general, mechanical refrigeration uses the low-temperature evaporation of a liquid to absorb heat in a set of coils. The resulting vapor is restored to its original liquid state by compressing and condensing it, abstracting the heat by passing water or air over a second set of coils at the outlet side of the compressor. Power for compression is usually supplied by an electric motor. The apparatus, exclusive of the evaporator or cooling coil, is known as a *condensing unit.* Two methods are available for applying mechanical refrigeration to unit conditioners.

The direct-expansion system provides for admitting the refrigerant through a pressure reducing (expansion) valve to the cooling coil, where its evaporation causes chilling of the surface over which the circulated air passes. Under this method, the equipment cost is low, the refrigerant lines need not be insulated, the apparatus is compact, and the operating expense is minimized by the avoidance of heat leakage and by the higher

permissible suction pressure at the compressor inlet (as compared with the indirect system). However, because of possible hazards from leaks, direct expansion is usually prohibited in hospitals and places of public assembly.

The indirect-expansion system uses a water-submerged coil in a tank near the condensing unit, for evaporation of the refrigerant. The chilled water or brine is then delivered under pressure by a motor-driven pump for distribution to the cooling coils in the individual unit conditioners, returning again to the tank. This avoids the possibility of refrigerant vapors, whether toxic or not, leaking into the conditioned rooms. Code

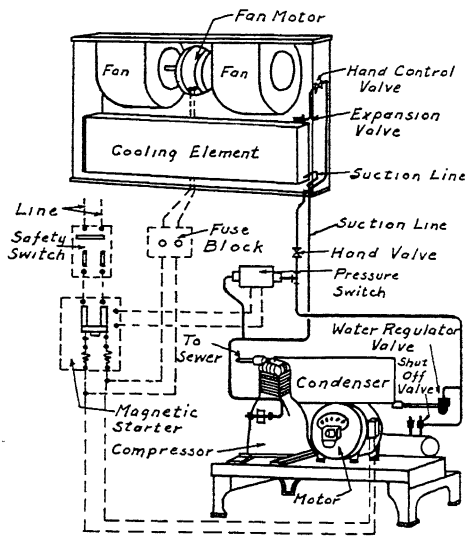

FIG. 6. ROOM COOLING UNIT

limitations on the quantity of refrigerant in the air conditioning apparatus are overcome, and a central condensing unit may be made to serve rooms on different floors or in remote parts of a building, without violating safety regulations. Difficulties that occur with compressor operation at less than 50 per cent of rated capacity are avoided through the use of a thermostat that shuts down the compressor when the tank water temperature reaches the set minimum; operation is had at constant suction pressure, independent of the number of unit conditioners running. With proper choice of temperatures at which the compressor starts and stops under thermostatic control, there is less *cycling* than with the direct expansion system. Under favorable conditions, the cooling coil may be supplied with steam or hot water for winter heating, thereby simplifying the construction of the unit conditioner, although at the expense of some complication in valved connections. However, the cold water tank and circulating pump take up room, and the cost of suitably insulated distribution piping is greater than that of equivalent liquid lines and suction returns for a direct-expansion system.

Separate Condensing Units

The separate condensing unit, for mechanical refrigeration with unit conditioners that are not self-contained, comprises the assembly, on a bedplate, of a compactly arranged compressor with motor, drive, condenser, liquid receiver, and automatic controls. The cylinder jacket of the compressor and the condenser may be cooled with water under pressure, or with air supplied by a fan mounted integrally with the compressor. A condensing unit connected to a single unit conditioner is shown in Fig. 6.

Steam-Jet Apparatus

Steam-jet (vacuum) refrigeration may be used in localities served by district steam mains, or in buildings with boiler plants available for summer use. While avoiding power-driven compressors, the steam-jet apparatus requires an appreciable amount of power for auxiliary pumps, and an increased quantity of cooling water to absorb the heat from the motive steam in addition to that abstracted from the conditioned rooms. Most installations of this type are of large capacity—above 20 tons refrigeration—but recently developed equipment is available for installations as small as 2 to 5 tons.

City or Well Water

Systems installed near the Great Lakes or in other regions where low cost cooling water is available in summer may often use this water directly in the coils of air conditioning units. In certain other places, well water can be obtained in sufficient quantity at moderate pumping expense. Restrictions on bulk use of water, or on discharge of large volumes into the sanitary sewers may prevent direct cooling.

Ice

Two methods of using ice are applicable: direct, with air circulated by a fan over ice cakes in an insulated tank within the room served; indirect, with an ice-melting tank remote from the unit conditioners, circulating chilled water to coils in the units by means of a motor-driven pump. The direct method has been employed with portable room coolers for hotel guest rooms, hospitals, and residences, where the demand for air conditioning is moderate and variable with respect to rooms served from day to day, and where it is feasible to move the units into a service room or kitchen for emptying and icing. The indirect method is identical with that common in theaters using ice, except that the water after spraying over the ice is pumped to unit conditioners instead of to a central fan system.

SOURCES OF HEAT

Steam or Hot Water Coils

The heating coils of unit conditioners for all-year or winter service are available for either steam or hot water, supplied at low or high pressure from building heating plants. Because the relatively high Btu per hour

outputs for heating (usually 1.5 to 3 times the rate for cooling), under thermostatic control, may produce disturbances in small heating systems, it is usually necessary that two-pipe steam systems operate at all times above atmospheric pressure, and that hot water systems have forced circulation with a pump. Unless the room space occupied by radiators in an existing building is needed for the unit conditioners or for other purposes, it is preferable that some or all of them be retained, so that the unit conditioners need supply only sufficient heat to permit their satisfactory operation for humidification.

Electric Elements or Gas Burners

Where energy is available at low cost, electric heating elements may be used in place of steam or hot water coils for winter service. Evaporation of water for winter humidification may likewise be accomplished electrically. More uniform control of temperature and humidity is practicable with electricity, because the heating elements may be divided into sections separately connected through thermostatically controlled switches. However, wiring connections must be larger than needed for summer conditioning with a compressor built into the unit; for instance, the power for a unit rated at 24,000 Btu per hour for winter heating is about seven times that used for 12,000 Btu per hour of summer cooling by the same unit.

A new unit conditioner employing the adsorption method for summer dehumidification is fitted with gas burners for winter heating. A part of the humidification is supplied by the water vapor resulting from combustion of hydrogen in the gas fuel, and the remainder by evaporation from a heated-water receptacle.

TYPES AND LOCATIONS OF UNIT CONDITIONERS

Fixed

The majority of unit conditioners are designed for floor mounting, preferably under windows. However, when radiators for winter heating occupy the window space and it is not desired to shift them or to eliminate them by using all-year type floor units, the location may be against interior partitions or alongside permanently situated furniture. In all cases care must be taken to insure that the direction of the air discharge will not cause drafts that may be objectionable to occupants. When outdoor air for ventilation is taken through the unit, the under-window position is advantageous, since it permits using a short inlet duct from louvers in a filler panel permanently inserted beneath the raised lower sash.

Ceiling or wall-mounted units may be used in commercial establishments, when floor space is at a premium. They generally secure refrigeration from a remotely placed condensing unit and are designed for support by means of hanger rods. It is often possible to conceal them in adjoining closets or workrooms, with the air discharge louvers fixed in the intervening wall; this makes it easy also to conceal the piping connections and wiring. In stores, suspended type units may conveniently be placed over the housed-in show windows.

Portable

Portable summer-function units mounted on rubber-tired casters or rollers can be obtained in the smaller capacities up to about 9000 Btu per hour. They may have built-in compressor units using city water for jacket and condenser cooling, or they may employ ice or low temperature city water. Hose connections for water supply and return, and for condensate drains, are needed in addition to a plug-in electrical connection. It is expected that a market for such units can be developed in hospitals, hotel guest rooms, and residences.

Special Types of Units

The field for unit conditioners has been extended by the appearance of small low-cost devices for comfort cooling that localize the cooling effect to the immediate vicinity of the user. These include bed tents and robe-type coolers, which require motors not larger than ½ hp. The tent is suspended from a bracket attached to the bed frame, and the cooler placed alongside is connected to it with a short collar for the air discharge. The robe-type device is intended for barber and beauty shops; it works on the same principle. Besides handling much smaller quantities of air, these expedients achieve economy because they operate only when required for the comfort of the user.

Reversed Refrigeration Heating

All-year unit conditioners that utilize their refrigeration apparatus for winter heating by the principle known as *reverse refrigeration cycle*, are being developed. A detailed explanation of this system is given in Chapter 39. For regions rarely having winter temperatures below freezing, there is believed to be a considerable field of application for such equipment. The heat delivered to the room will range between 2.5 and 3.5 times the equivalent of the electrical power taken by the motor, depending on the outdoor temperature. The gain is, of course, derived from the ambient air, requiring an inlet and an outlet duct for passing a considerable volume. Lower rates for energy may sometimes be obtained, under the resulting improved annual load factor, when both cooling and heating are provided electrically.

LOCATION OF UNITS, AIR FLOW PATHS

The number of units, the availability of space, and the convenience of making piping, wiring, and duct connections, which involves the location of outside cooling, heating, and power sources, must be considered in choosing locations for the units, as must the positions of persons, furniture, and materials in the space to be conditioned, and the requirements of air distribution.

The most important of these considerations is air distribution, and units should be so located as to secure uniformity in all parts of the room whether the application is for comfort conditioning or for industrial uses. The discharge of cooled air, in general, should be upward immediately at the conditioner, with sufficient horizontal component to carry to the most remote point; return to the inlet of the unit, which occurs below the

breathing line and along the floor, should be at low velocity. The location of doorways, air vents, and sources of heat should be studied, as they have a marked effect on air flow and on temperature uniformity. Infiltration through leaky windows with certain wind directions likewise disturbs or restricts the circulation of air from the unit conditioner, and frequently causes cold spots by preventing diffusion at the ceiling. Velocities below the breathing line should be kept low—not over 40 to 70 feet per minute; in this range, an anemometer will not work, and the Kata thermometer must be used for testing purposes.

CONSTRUCTION OF APPARATUS

Description of Typical Units

The types and designs of air conditioning units proposed or in production are legion; new designs are constantly appearing, with a tendency toward better mechanical construction and a wider range of application. However, nearly all types now commercially available utilize mechanical refrigeration or cold water for summer cooling, and consequently the descriptions below are limited to such equipment, using electric power. Illustrations of current makes and models will be found in the *Catalog Data Section* of this volume.

Fig. 7 shows an all-year, floor-type unit for direct expansion of refrigerant supplied by a remotely located compressor; with modifications, the cooling coil can be used with chilled water. The fans below the separate cooling and heating elements deliver the air against deflectors that give distribution across the element face, and the usual drip pan for condensation is provided. Separate elements for heating and for cooling possess the advantage of allowing the former to be connected to the source of heat with piping entirely separate from the refrigerant lines to the cooling element, with no cross-connections. Thus the unit may be used for warming in the morning and for cooling later in the day, if desired, without manipulation of valves. When this unit is installed for cooling only, the heating element is omitted.

A summer-function unit with fans above the cooling element is shown in Fig. 6; a condensing unit, with schematic diagram of refrigerant piping and wiring, is included. This air conditioning unit, as well as that in Fig. 7, when housed in an ornamental cabinet, is suitable for high grade residential or commercial installations.

An entirely different arrangement, shown in Fig. 8, places both the air inlet and the discharge at the top of the unit. The fan in the upper portion at one side discharges the air toward the bottom, where it turns and passes horizontally through an air washer equipped with atomizing sprays. The path continues vertically upward through eliminators, cooling surface, and heating surface before leaving the unit. With steam or hot water connected to the heating element, tempered water to the sprays, and refrigerated water to the cooling element, this unit gives controlled temperature, humidity, air cleaning and air movement in both summer and winter. Air washing may be continued in summer to remove room odors. Acoustical treatment of the housing and the outlet baffles permits installation where the noise requirements are exacting.

One of the most recently developed units, designed particularly for low

cost installations, is shown in Fig.9. The twin fans with wheels, mounted on extensions of the motor shaft, take air from the floor and send it downward through a passage containing a water spray. The direction of flow is then reversed, the air passing through a double set of coils for cooling or heating, and leaving the cabinet through a top grille. The

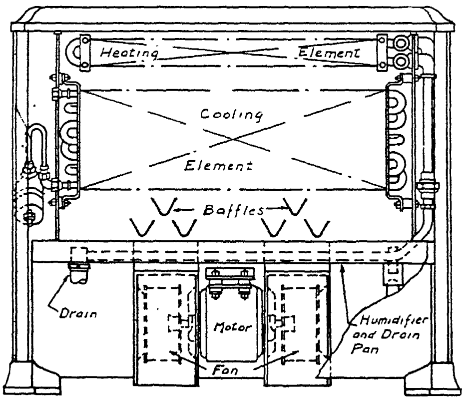

FIG. 7. FLOOR UNIT FOR HEATING AND COOLING

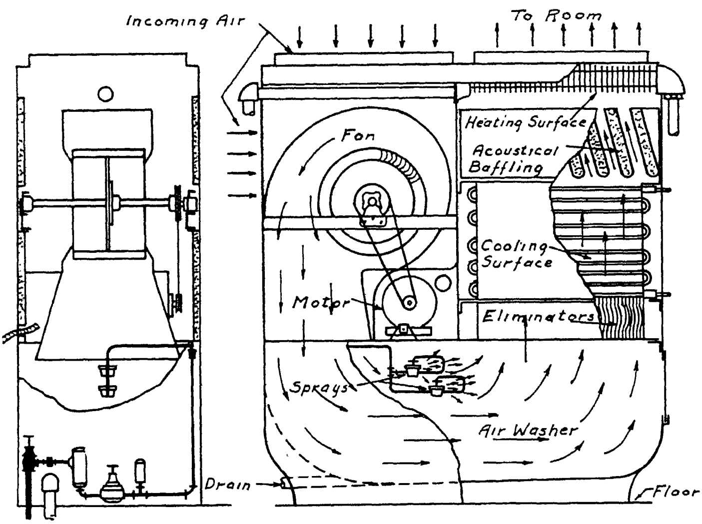

FIG. 8. UNIT WITH TOP INLET AND OUTLET

spray nozzles are supplied with city water, the excess collecting in the air reversal chamber, which has a drain. The cooling coil uses water from the city mains or other low-temperature source; alternatively, direct expansion of refrigerant from a motor-driven condensing unit can be utilized. The unit provides all-year functions, the cleaning being accom-

plished by the spray in winter and by contact with the wetted cooling coil in summer. Automatic electrically operated controls for water flow, steam flow, temperature, and humidity are optional.

For industrial applications, the floor-type unit, Fig. 10, or the ceiling-type, Fig. 11, may be used. The former has a galvanized steel casing that encloses the cooling and heating elements, with fans mounted above them; the air discharges from the top through 90-degree elbow ducts, which deliver it in a nearly horizontal direction. The operating motor for the fan is carried on a bracket at one side, and at the bottom a condensate drip pan is provided; space between the pan and motor bracket is utilized

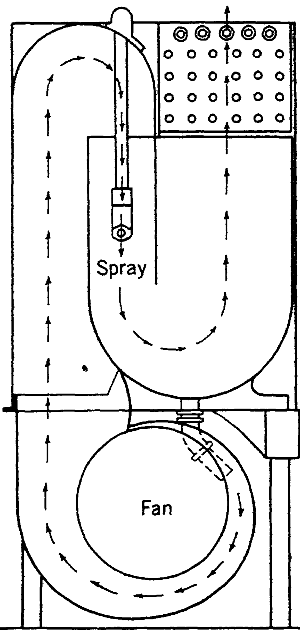

FIG. 9. ALL-YEAR TYPE UNIT CONDITIONER

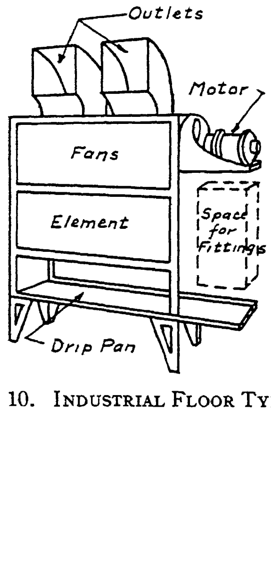

FIG. 10. INDUSTRIAL FLOOR TYPE

for traps and valves. This unit does not wash or filter the air, nor is a fresh-air supply provided for ventilation; thus only cooling, dehumidification, and circulation can be accomplished in summer, and heating and circulation in winter.

The ceiling type, Fig. 11, is primarily for summer use, although when supplemented by a regular heating system it can accomplish a limited amount of humidifying in winter. The apparatus consists of an air washer with the usual water sprays, eliminator plates, and air circulating fan, designed for suspension from the ceiling. The air supply is taken from the room through the intake register, passes through the water spray and eliminators, and is delivered back into the room through the discharge outlet equipped with adjustable louvers. The refrigeration unit may be located at any convenient point and the cooled water circulated to and

from the conditioner through pipes at the ceiling, so that no floor space is lost. This style of unit is for industrial and large office installations. Where the appearance on the ceiling is objectionable, the unit may be placed at some other location, using a compact duct system for the air to and from the conditioning unit.

In the following paragraphs, typical forms of construction are outlined. Many variations of these have been used, and modifications or entirely new details are constantly being introduced. The *Catalog Data Section* illustrates and describes the current designs of leading equipment manufacturers.

Cabinets, Registers

Cabinets are made of *furniture grade* sheet steel suitable for pressing in panels, protected by corrosion-resistant priming coatings. The design is such as to permit access to the equipment, which is independently supported on a frame or chassis. Heat insulation of either rigid or flexible

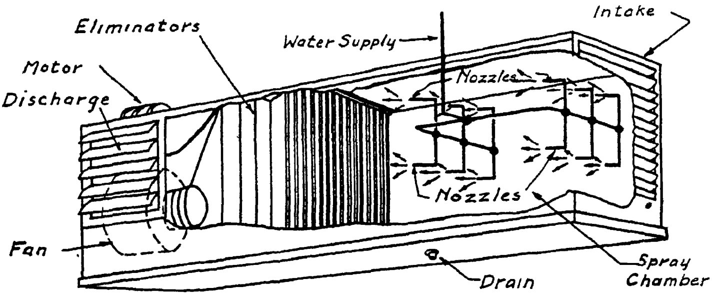

FIG. 11. SUMMER COOLING UNIT

type, to prevent sweating in summer or overheating in winter, is used, particularly with thermostatic controls that start and stop the fans without affecting the supply of heating or cooling medium to the coils. Sound-deadening is equally important, to avoid vibration or drumming effect of the panels. The finish of commercial and residential units is usually in imitation of wood grain, or may be in solid color to harmonize with room finish and furnishings.

Outlet registers are generally placed in the top of the cabinet to direct the air at an angle approximately 30 degrees from the vertical. They should be proportioned to maintain sufficiently high air velocity for preventing a local cold spot caused by too short a flow circuit in the room. Types that give ejector action, entraining some room air and propelling the mixture a considerable distance away from the unit, are preferred. Return-air registers should act as sound-deadeners and serve to hide the internal mechanism.

Motors

Motors are usually of the capacitor or repulsion-induction types, single-phase. However, in sizes 5 hp and larger, three-phase will ordinarily be preferable; this will depend on character and capacity of service facilities

available. Special designs giving low starting current, silent running and (in the case of compressor motors) high starting torque, are essential. Features to minimize lamp flicker and radio interference must be incorporated. coördinated with characteristics of the compressor. For automatically controlled units, two motors (with sequence relay for consecutive starting) are sometimes direct-connected to the load, for holding the current inrush to a low value, when the starting torque of the driven equipment permits. Devices known as *suction unloaders*, permitting large air or refrigeration compressors to come up to speed without load, involve too much complication for the size of apparatus used with unit conditioners.

Refrigerants

The choice of refrigerants for a direct-expansion system is limited to non-toxic, nearly odorless fluids—principally methyl chloride, freon or iso-butane. Local ordinances and fire regulations prescribe the maximum quantity of refrigerant in a system for residential and usual commercial requirements. Indirect systems may use ammonia, sulphur dioxide or carbon dioxide, since the equipment and piping can then be isolated, remote from the conditioned rooms.

Compressors, Condensers, Cooling Coils or Evaporators

Compressors of the multi-cylinder reciprocating or rotary designs are preferred, as they minimize starting troubles and lamp flicker. Gland or shaft-seal leaks, with freon or methyl chloride, must be provided against, because of the difficulty in detecting leaks before the refrigerant charge is lost; this is especially important when the pressure in the crankcase tends to rise after the compressor shuts down. V-belt drives from motors permit the compressor and motor each to run at its most economical speed, and provide desirable resilience at the instant of starting.

Condensers for water cooling are of either the double-tube or shell-and-tube types, with the latter preferred when the water carries dissolved or suspended solids; provision for opening and cleaning should be made. Air cooled condensers usually are supplied with air by propeller fans integral with the compressor flywheels or mounted on the compressor shafts.

Evaporator coils, in units using direct expansion of the refrigerant, also constitute the cooling coils over which the air flows to be cooled and dehumidified. They are constructed of metal suitable for the refrigerant used, and have fins on the exterior to increase the heat transfer per unit length of tube. The arrangement and amount of surface provided, in relation to the maintained refrigerant temperature, the initial temperature and dew point of the air, and the rate of air circulation over the coil determine the final air temperature and thus the amount of dehumidification secured. With indirect refrigerating systems, the cooling coils in the units are usually somewhat larger, because the cooling fluid (water or brine) is at a higher inlet temperature; the evaporator in this case is remotely located (with the condensing unit) and serves to chill the water or brine circulated by a pump to the coils in the unit conditioners.

Collection of water condensed on cooling coils requires a drip pan, with drain piped to a disposal point often fixed by sanitary regulations, or an

ejector operated from the city water supply used for winter humidification or for cooling the condenser. In some cases, a condensate storage tank to be emptied manually, or a motor-driven pump, is supplied. Eliminator baffles may be provided immediately below the outlet grille to intercept any drops of water picked up by the air current.

Humidifiers

Humidification in unit conditioners may be accomplished by sprays using cold or heated water at city main pressure, or by water trickling over heated surfaces or a mesh filling. The design must provide for supplying the heat of evaporation, and for exposing to the air current a sufficient area of water film. This requires a considerable excess of water, which may be wasted to a drain or recirculated by a pump; with the latter, periodic flushing of the system must be practicable, to dispose of the dirt removed from the air. When considerable amounts of fresh air are provided by the unit conditioner or enter by infiltration, the quantity of water to maintain the desired humidity is greater than when the unit merely recirculates and the room has only moderate leakage. In the latter case the humidifier can be small, since only a slight amount of moisture is supplied to the air with each passage through the unit.

Fans, Fresh Air Supply

Fans are usually of the centrifugal type with scrolls, inlet cones, blades, and tip speeds designed for quiet operation. Compactness and uniform distribution of air across the width of the coils and grilles are obtained by using two or more fans in parallel, the rotors mounted on a common shaft or on a double-end extension of the motor shaft. Housings and deflectors (if used at the fan outlets) may be acoustically treated. Propeller-type fans are sometimes used, although more difficult to make quiet. Efficiency is a secondary consideration, because the motors are of small fractional-horsepower sizes.

Fresh air supply connections are usually through a fixed panel inserted in a window frame between sill and lower sash; this has a louvered and screened opening, connected with a metal duct to the space in the cabinet at the inlet side of the fans. A manually adjustable damper regulates the proportionate volumes of recirculated and fresh air.

Filters

Air cleaning devices include filters of glass or metal wool, cellulose, felt, or woven fabric; they are usually of the renewable cartridge type, designed for low air resistance. Types especially effective in the removal of hay fever pollen are desirable. An alternative device is a water spray also serving as a humidifier in winter, or when supplied with chilled water, as a cooler and dehumidifier in summer. The fins on cooling coils, automatically wetted by condensate obtained in dehumidifying the air, are also employed in some types. Complete removal of tobacco smoke is not possible with any type of filter or washer used in unit conditioners; the limited amount of ventilation air in summer, admissible from the operating cost standpoint, often results in a smoke haze. The only remedy is increased ventilation, with consequent higher operating expense; the air

outlets should be near the ceiling, to tap the upper stratum where the smoke is most dense.

Heating Coils

Heating coils are generally of the extended-surface type for compactness, and may be designed for any pressure of steam available, or for forced circulation hot water—usually about 180 F. Some types of heating systems, both hot water and steam, are unsuited to unit conditioners, especially when thermostatically controlled, because of the resulting sudden changes in load. Gravity-circulation hot water systems must be converted to forced circulation, if unit conditioners are to be connected, and steam systems must always operate at pressures above atmosphere.

Manual and Automatic Controls

Manual control is generally used with unit conditioners, because of the high cost of reliable automatic controls. Fan and compressor motors are started and stopped by individual switches. Fluids for the heating and cooling coils are regulated with manual valves, generally permitting the flow to continue regardless of whether the fan is operating; with this arrangement, adequate heat insulation must be provided within the cabinet, and the size of the unit conditioner in a room is limited to that which will give the minimum required heat supply by gravity air circulation through the conditioner when the fan is stopped.

Automatic controls consist of a thermostat for room temperature and a hygrostat for humidity. The former starts and stops the fan in the unit conditioner, thereby controlling the supply of cooled or warmed air to the room. A hygrostat is not usually supplied, because of high cost and imperfect reliability of types now available; when used, it is connected to the valve admitting water to a spray- or trickle-type humidifier, or to a refrigerant supply valve controlling a supplementary section of the cooling coil. The best arrangement is one that permits the full capacity of the compressor to be utilized for either sensible heat removal or dehumidification, based on the principle that the compressor capacity varies approximately as the temperature of the refrigerant in the cooling coil. Compressors are started and stopped by pressure switches on the discharge (high pressure) side. Water supply to the compressor jackets and the compressor is turned on and off by a solenoid valve energized when the compressor motor starts. Refrigerant supply to the cooling coil (constituting the evaporator) is usually regulated by a thermostatic valve, as a function of the refrigerant outlet temperature, or by a flow valve that tends to hold a constant level in the liquid receiver.

INSTALLATION OF UNIT CONDITIONERS

Piping, Wiring, Ducts

Piping connections for water and steam are made preferably with corrosion resisting material, usually brass or copper. Light weight rigid tubing with sweated joint fittings has advantages over threaded construction. Flexible copper tubing with compression type connections may be used in the smaller sizes (up to ¾ in. dia.), as it lends itself to concealment in existing walls or other places difficult of access; distribution of the

refrigerant from a remotely located condensing unit is usually made with such flexible tubing.

Wiring connections should be made using modern materials and methods, such as will receive approval of local inspection authorities having jurisdiction. For portable conditioning units with built-in compressors, particular care should be taken to select rugged receptacles and plugs; waterproof flexible cords are recommended because of the possibility of water leakage from adjacent hose connections or by overflow from the unit if the drain becomes stopped.

Ducts for outgoing air supply, usually from nearby window openings, present no particular problems.

Workmanship

The requirements as to workmanship for installation of unit conditioners are exceptionally severe; this is particularly true for work in high grade offices and residences, in occupied quarters. Handling of the materials and the cutting, patching, and refinishing invariably demand neatness, accuracy, and planning that the ordinary mechanic is unfamiliar with, so that close supervision must be given.

BASIS OF EQUIPMENT RATINGS[1]

While no uniform standard for rating unit air conditioners has yet been adopted, manufacturers generally give a definite rating for each size unit, based on the volume of air handled by the fan for cooling; the rating is stated in Btu per hour at a given dry- and wet-bulb temperature of air entering the unit, with a given refrigerant temperature maintained within the coil, resulting in a stated relationship between sensible and latent heat removal. The temperature of the cooling water or air supply for the condensing unit is also involved. The duty for heating service is likewise given in Btu per hour with 70 F room temperature, for a stated steam pressure or hot water temperature (usually 180 F). Humidifying capacity is based on hourly weight of water evaporated. The *Catalog Data Section* in this volume gives the ratings of current models offered by leading manufacturers.

METHODS OF CALCULATING REQUIRED CAPACITY

In estimating the load for unit air conditioning apparatus, a survey should be made of the surrounding conditions and the heat quantities calculated. The climatic conditions representing the maximum loads to be designed for should be carefully determined.

Cooling Loads

For cooling loads served by unit conditioners, the factors for heat gains and losses are the same as apply to central fan systems. The sensible heat gains are from the following sources:

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[1]Refer to the standard ratings of air conditioning equipment of the *National Electric Manufacturers Association*.

a. Sun effect.
b. Transmission through walls, floors, ceilings, glass, and roofs.
c. Infiltration, including ventilation air.
d. People.
e. Lights.
f. Electric motors and appliances.
g. Steam and gas appliances.
h. Miscellaneous heat sources.

The latent heat load, usually determined separately, comes from dehumidification of the air and from people and materials. The method and factors to be used are outlined in standard texts and in manufacturers' handbooks.

Rated capacity for direct-expansion refrigeration units should include an allowance for the heat equivalent of the fan-motor input, plus the portion of the power to the compressor not removed by the cooling water. For indirect-expansion systems, allowance should be made for heat pickup by the refrigerant circulating lines, or for the pickup by a chilled-water or brine-storage tank and for the shaft-horsepower input to a circulating pump, if used.

As a rough approximation, the refrigeration tonnage required for unit conditioners serving rooms devoted to various uses may be assumed as follows:

Types of Rooms	Cu Ft per Ton
Cafeterias, lunchrooms	1000 to 1500
Barber and beauty shops, dance halls	1200 to 1800
Dining rooms, crowded retail stores	1500 to 2000
Theaters	1800 to 2400
General offices, club rooms, retail stores, funeral parlors	2000 to 3000
Banks, brokers' offices, private offices, residences	2500 to 4000

Obviously, there will be many cases to which the mentioned limiting values do not apply. A calculation of the cooling load, based on an accurate survey, should always be made before recommending the size of an installation or naming a cost figure.

Heating Loads

Heating loads are calculated in the usual manner, as outlined in Chapter 7. Allowance must be made also for the latent heat supplied to the water for humidification, when the infiltration or ventilation air quantity is large.

APPROXIMATE COSTS

Equipment and Installation

Floor type all-year unit conditioners, completely self-contained and equipped with motor-driven compressors and thermostatic controls, rated at about 12,000 Btu per hour for cooling, cost $500 or more at the factory. To this must be added freight and installation charges, in-

cluding expense for piping, wiring and fresh-air ducts, amounting to between $75 and $150, with perhaps $50 additional if overtime work is necessary to avoid inconveniencing the occupants of offices or other quarters.

A similar unit without compressor, using chilled water or direct-expansion refrigerant from a remotely located compressor, costs $175 or more at the factory. Installation expense is somewhat greater than for the self-contained unit, because the refrigerant piping costs more than is saved by the reduction in wiring. Omission of the heating coil, confining the unit to summer functions only, lowers the price by $25 to $100.

For smaller units, rated between 6000 and 8000 Btu per hour, providing all-year service and equipped with motor-driven compressors, the price ranges from $325 to $450 at the factory. Larger units, rated at about 24,000 Btu per hour for cooling, cost between 25 and 45 per cent more than the 12,000-Btu per hour size. Delivery and installation expense for either of these sizes does not differ more than 25 per cent from that of the 12,000-Btu per hour unit.

Industrial-type conditioners, either floor or suspended models, are usually made only in ratings of 20,000 Btu per hour and higher; omission of expensively finished cabinets and other differences reduces the cost considerably below that of corresponding sizes of commercial and residential types.

Condensing units completely assembled on bedplates, especially adapted to serve one or more unit conditioners, are available. They comprise a motor, compressor, condenser, liquid receiver, and control devices, and they are arranged for water cooling or are equipped (in the smaller sizes) with fans for air cooling. Prices for representative sizes, including motors but not starting equipment, are as follows:

Btu per Hour	Factory Price	Installation Cost
8,000	$275 and up	$60 and up
12,000	325 and up	65 and up
36,000	575 and up	80 and up
60,000	800 and up	90 and up
120,000	1100 and up	125 and up

These prices are for water-cooled types; air cooling adds $25 to $50. Installation cost includes transportation, foundations, wiring, starting equipment, cooling water piping or air ducts, and sound-deadening insulation. Refrigerant connections from liquid receiver and compressor suction to unit conditioners are not included. For office buildings and similar occupied quarters, overtime labor may increase the cost.

The prices given represent net cost to the ultimate purchaser. Although roughly indicative of the present-day market, they should not be used as a basis of a specific estimate or an appropriation, because designs, ratings, and prices vary considerably between makers and in different parts of the country. Transportation and installation expense is even more variable, depending upon freight rates, wage scales, and particularly on the condition of the building and the adequacy of existing piping and wiring systems to which the unit conditioners are to be connected. Furthermore,

the industry is in a state of fairly rapid development, so that any general cost figures should be used with caution.

Operation

For a 24,000-Btu per hour unit operating at full load for summer cooling, with electricity at $0.05 per kwh and 70 F city water at $1.50 per 1000 cu ft, the hourly electric and water expense works out to $0.14. Under climatic conditions representative of a large part of the country, the load factor, during the 10 hours' daily operation required, averages 50 per cent; this gives a daily operating cost of $0.70. The seasonal cost for localities requiring, for example, 1000 hours of operation (at 50 per cent load factor) then becomes $70. To this should be added maintenance and fixed charges of 25 per cent (based on about a five-year useful life) on an investment around $1200. The over-all expense for owning and operating is thus of the order of $370 per year. Such a cost may be incurred, in a climate like that of New York City, by the owner of a home in which at least the living room, the dining room, and a bedroom are cooled with unit conditioners served by refrigerating equipment of the mentioned capacity.

This expense may be compared with the cost of winter heating, computed by adding annual fixed and maintenance charges to cost of fuel, attendance, and other items. The comfort attainable in hot, humid weather is so welcome that these costs will undoubtedly be looked upon as reasonable by an increasing number of people, as they become personally familiar with the value of the service rendered by modern air conditioning equipment. Exposure to such comfort in commercial establishments, railroad trains, and other public places will unquestionably tend to increase the demand for home installations at a greater rate each year. The developments in equipment for the type of service described have been rapid during the past few years and the latest models may be seen in the *Catalog Data Section*.

PROBLEMS IN PRACTICE

1 ● Are unit conditioners necessarily self-contained?

No. The heating medium is always supplied from a separate plant, and the refrigerant for cooling and dehumidification may come from a separately located compressor or other supply source.

2 ● Are ducts used with unit conditioners?

Yes. Usually a short connection for fresh air intake is made to an adjacent window or wall opening. Occasionally ducts are required for return air and for discharge, when a unit is located near the room served but not within it.

3 ● What is the meaning of the term condensing unit in relation to unit air conditioners?

A condensing unit is the assembly, on a bedplate, of a compactly arranged refrigeration compressor, motor, drive, condenser, liquid receiver, and automatic controls used for supplying the refrigeration.

4 ● Why are metal surface cooling elements instead of liquid spray chambers used in the design of most unit air conditioners and unit coolers?

The first cost of the surface cooling type of unit is considerably less than the cost of spray type equipment. Further, the requirements of many industrial air conditioning jobs and of all comfort cooling jobs where unit equipment is applicable can often be effectively met with the use of surface type units, with a reduction in the space required for making the installation. Where space conditions are especially limited, the cross-sectional area of the surface cooler can be reduced because the resulting increase in velocity over the coil surface increases the effectiveness of the surface, whereas an increase in velocity through a liquid spray would reduce its effectiveness.

5 ● Why are air conditioning units with metal cooling surfaces not desirable for all industrial jobs?

Wherever unusually close control of relative humidity is required, a spray type unit will prove to be more satisfactory. Relative humidity control and accurate temperature control, however, can be maintained without difficulty with the use of metal surface units.

6 ● Why is accurate control of relative humidity with surface coolers more or less complicated?

A surface cooler cannot add moisture to the air, and moisture is removed only when the surface temperature is below the entering dew-point temperature. Any change in condition of the entering air will result in a change in the dry-bulb depression of the leaving air. This change in entering condition requires not only a readjustment of the air volume but also a change in the coil temperature, if accurate control over the relative humidity is to be maintained.

7 ● What in general are the characteristics of unit conditioner operation using surface coils?

For a constant entering dry-bulb temperature and a constant refrigerant temperature any increase in the entering wet-bulb temperature will produce a rise in the leaving dry-bulb temperature with an accompanying reduction in the wet-bulb depression of the leaving air. The sensible heat removed by the unit decreases and the latent heat increases, while the total heat removed also increases. When the dry-bulb temperature of entering air is increased, with constant refrigerant temperature and constant wet-bulb temperature of entering air, the wet-bulb depression of the leaving air increases, and since it is this depression which determines the maintained relative humidity it must be carefully considered when selecting the unit.

8 ● If a drop in the dry-bulb temperature of entering air reduces the capacity of the unit, is there not danger of selecting a unit which is too small, if its selection should be based on an excessive entering dry-bulb temperature?

Yes. If the total cooling load is largely internal (such as from occupants and lights) as distinguished from the cooling load of outdoor air, and the unit is selected on the basis of a too high dry-bulb temperature of entering air, then, in the event of under capacity, it might be possible to maintain the room temperature by reducing the quantity of outdoor air. But this increases the recirculated air taken into the unit, reducing the dry-bulb temperature of entering air and, therefore, reducing the sensible heat capacity of the unit. This reduction in capacity may offset the gain obtained by reducing the amount of outdoor air taken in. Further, since the total tonnage required for any installation is equal to the total internal heat load plus the total heat removed from the outdoor air, and since the outdoor air might have a wet-bulb temperature equal to the designed wet-bulb but less than the designed dry-bulb temperature, then the sensible heat capacity of the unit will be less than that required. It follows that unit air conditioners and coolers should not be selected on a basis of the maximum possible dry-bulb temperature of entering air.

Chapter 13

UNIT HEATERS, VENTILATORS, AND COOLERS

Types of Unit Heaters, Heating Media, Entering and Delivery Temperature, Output of Unit Heaters, Direction of Discharge, Boiler Capacity, Direct-Fired Units, Unit Ventilators, Split and Combined Systems, Location of Unit Ventilators, Capacities, Attic Fans, Unit Coolers

A UNIT heater consists of the combination of a heating element and a fan or blower having a common enclosure, and placed within or adjacent to the space to be heated. Generally, no ducts are attached to the inlets or outlets. A unit ventilator is similar in principle of operation to a unit heater, but is designed to use all or part outdoor air with or without alternate provision for handling recirculated air. Unit heaters are designed mainly for factory and industrial use, whereas unit ventilators are intended largely for school and office ventilation and heating.

Unit heaters and unit ventilators are designed to:

1. Circulate the air in the building at a rapid rate.
2. Reduce the temperature differential between floor and ceiling.
3. Direct the heated air so as to accomplish the positive and rapid placing of the heat where it is effective.
4. Remove the cold stratum of air from the floor.

TYPES OF UNITS

There are many types of unit heaters available. Most of them employ convectors to be supplied with steam or hot water. Some are mounted on the floor, whereas others are designed for suspension overhead. Heating surfaces in the form of steel pipe coils, non-ferrous tubes or shapes with extended surfaces, cast-iron, and pressed and built-up sections of the cartridge or automotive type are all used in unit heater construction.

Among the unit heaters available are types designed especially for industrial purposes having from one to four warm air outlets per heater which may be arranged to discharge in selected directions and which will project their heating effects over distances of from 30 to 200 ft from the heater, depending upon the capacity of the heater and upon the design of the fan and outlets. Because these heaters have been satisfactory when placed as far as 400 ft from each other, it is possible to select the heater location best suited to the production layout in factories. There are available propeller fan type heaters of smaller capacity with outlet velocities of from 300 to 800 fpm, and these may be placed from 30 to 100 ft apart.

Table 1. Constants for Determining the Capacity of Blow-Through Type Unit Heaters for Various Steam Pressures and Temperatures of Entering Air

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	**1.000**	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.024	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

HEATING MEDIA

The convectors in unit heaters or ventilators may be supplied with either hot water or steam. When water is used, it should be circulated mechanically, and the pumpage rate and friction loss should be based upon test data from the particular unit to be employed. The heat output of a given heater will be less when using water than with steam, even at the same temperature.

Either high or low pressure steam may be used, but the proper venting of air and the prevention of flash steam from the condensate in the returns become increasingly troublesome as the steam pressure increases. The use of properly constructed traps with some reliable form of thermostatic air by-pass solves the first of these problems, while proper venting or the use of condensing legs solves the second. Increasing the return temperature tends to increase return line corrosion, especially at points where overheated condensate or steam are led into a line.

When low pressure steam is used with unit heaters and ventilating units it is highly important that proper means be provided for taking care of the heavy condensation. They should not be applied to low pressure gravity return systems except where the difference between the heater level and the boiler water line is large enough to compensate for the pressure loss through the convector at its highest possible condensation rate. The use of vacuum or return pumps and receivers is advisable, with jobs of any considerable size, as the surest way of taking care of condensate and at the same time providing for proper venting of the units directly into a vacuum return line system, or into an open vented return system, the latter having some advantage in preventing the formation of any vacuum in the unit itself, which sometimes tends to hold up condensate and cause freezing.

ESTIMATING HEAT LOSSES

The heat losses of a building to be equipped with unit heaters are determined in the same manner as for any other heating system, excepting so far as the unit heaters may prevent air stratification and thus reduce the temperature difference between the ceiling and floor. (See Chapter 7.)

Unit heaters may be arranged to recirculate the air or to supply warmed air from the outside for ventilation or to make up air exhausted.

If all or a part of the air is to be taken in from out-of-doors, the heat necessary to warm this air from the outside temperature to the inside temperature must be added to the transmission or other losses. Units of the number and size needed to furnish the total heat required are then selected from the manufacturers' rating tables, using these ratings at the steam pressure to be used and at the temperature at which the air will enter the convector.

AIR TEMPERATURES

For recirculating heaters with intakes at the floor level, the temperature to be maintained in the room should be used as the temperature of the air entering the heater. Where suspended heaters are used without any intake boxes extending down to the floor level, a higher entering air

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF Draw-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Air Entering Heater											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	**1.000**	0.930	0.861	0.792	0.728
5	1.565	1.485	1.410	1.334	1.260	1.187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1.078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

temperature should be used than that at which the room is to be maintained. With suspended heaters taking in air at some distance above the floor, the temperature variation from floor to ceiling may reach as much as 1 deg for each foot of elevation during periods when the maximum capacity of the heaters is required. Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. These temperature differences per foot of elevation are less than the corresponding variations per foot of elevation for spaces heated by direct radiation.

Unit heaters save fuel because of their ability to circulate air at a lower average temperature than the air circulated by direct radiators; however, the unit heaters must circulate more air in any given time than is needed with direct radiators. This requires the selection of heaters having a liberal air capacity for the required heat output, which in turn means a relatively low final temperature. Extremely low final temperatures can be had only at the expense of larger heaters and increased power, so that an economic limit is imposed. In general, for heating purposes it is advisable to use a delivery temperature not more than 70 F above the average room temperature desired, and one considerably less where possible.

OUTPUT OF HEATERS

It is standard practice to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[1]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heat capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Tables 1 and 2. Table 1 is for blow-through and Table 2 is for draw-through unit heaters. These tables are accurate within 5 per cent.

The ratings customarily published for unit heaters apply only for recirculation and free discharge, unless otherwise noted in the rating tables. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heat capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design, and speed of the fans employed, so that no specific percentage of reduction can be assigned for all heaters for a given added resistance. In general, however, disc or propeller fan units will have a larger reduction in capacity than housed fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

[1]See A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

When steam supplied to the heaters contains superheat, the capacity of the heater will be but slightly less than with saturated steam at the same pressure. Recent tests indicate that the reduction of capacity from this cause is negligible for superheat up to 50 deg and will not exceed 3½ per cent for any degree of superheat.

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter directions.

DIRECTION OF DISCHARGE

Various types and makes of unit heaters are illustrated in the *Catalog Section* of this edition. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

BOILER CAPACITY

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when

proper provision is made for returning the condensate. If ventilators are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

QUIETNESS

In selecting unit heaters, attention should be given to the degree of quietness required for the installation.

No given fan speed may be applied as a measure of relative quietness to fans of different designs and proportions. Quietness is a function of type, diameter, blade form and other variables besides speed, and all

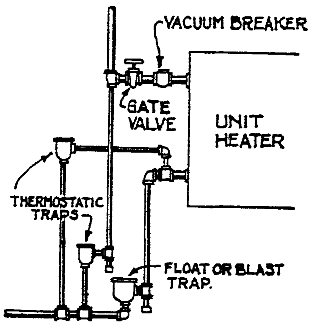


FIG. 1. UNIT HEATER CONNECTIONS WHERE CONDENSATION IS RETURNED TO VACUUM PUMP OR TO AN OPEN VENTED RECEIVER

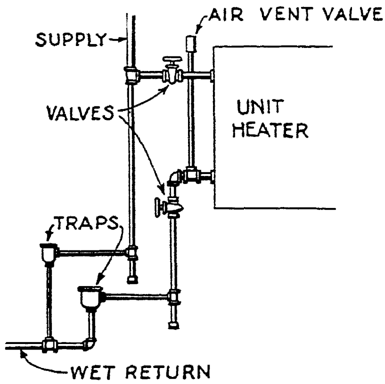


FIG. 2. UNIT HEATER CONNECTIONS WHERE CONDENSATION IS RETURNED TO BOILER THROUGH WET RETURN

these must be taken into account. In general small fans may be run at higher motor speeds than large fans with equal quietness; and centrifugal fans are more easily made quiet than disc or propeller fans.

PIPING CONNECTIONS

Piping connections for unit heaters are similar to those for other types of fan-blast heaters. Typical connections are shown in Figs. 1 and 2.

One-pipe gravity and vapor systems are not recommended for unit heater work.

For two-pipe closed gravity return systems the return from each unit should be fitted with a heavy-duty or blast trap, and an automatic air valve should be connected into the return header of each unit. Pressure-drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual air valves on the heaters, or it may be carried into the returns the same as for vacuum systems and the entire return system be free-vented to the

atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vented systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with some danger of return pipe corrosion and the problem of its elimination at some other point in the system.

The connections for steam and return piping to unit heaters must always be calculated on the basis of the high heat emission or condensation rate of such devices. The pipe-size tables given in Chapter 32 may be used for unit heater work by multiplying EDR values by 240 to get Btu values.

OTHER TYPES OF UNITS

All Electric

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. On rare occasions electrical resistances are used as the heating element. These are applied only where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. (See Chapter 39.)

Direct Fired

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm air type and are equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air in to the space to be heated. As is the case with the steam fed unit heaters, the gas fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified.

Turbine Driven

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

INDUSTRIAL USES

In addition to their prime function of heating buildings, unit heaters may be adapted to a number of industrial processes, such as drying

and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye-houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 7.)

Information on the control of unit heaters will be found in Chapter 14.

UNIT VENTILATORS[2]

A unit ventilator must be pleasing in design because it is generally used where it must harmonize with the furniture or with the decorative scheme. It consists usually of a rectangular steel cabinet finished with an enameled surface and containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.
3. Adhesive or dry type filters for cleaning the air (optional).
4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together. (No mixing chamber is normally provided where sectional type convectors are used.)
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Device for ozonizing air (optional).
9. Discharge grille or diffuser.
10. Temperature control arrangement.

The primary functions of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air.
2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.
3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating.
4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.
5. To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary.
6. To perform all its functions without objectionable noise.

In addition to these functions, unit ventilators frequently are arranged so that the air supplied may be cleaned by means of filters of either the dry or viscous type. If filters are used, the proper allowance must be made for the increased resistance offered to the air flow. Humidifiers in unit ventilators are rather difficult to control and are only furnished upon special order.

[2]A roof ventilator is sometimes termed a *unit ventilator*. For information on roof ventilators, see Chapter 4.

1. *Air Supply for Ventilation.* The outdoor air supply for ventilation is delivered by motor-driven fans operated at comparatively low speeds, the back of the cabinet being connected to the outside through rust-proof louvers and screens. Air quantities may be estimated on the basis of data given in Chapter 2. (See A.S.H.V.E. Ventilation Standards.)

2. *Warming Incoming Air.* The air is heated by passing it through specially designed convectors. The amount of heating surface to be provided in the unit is determined by the volume of air to be heated and the temperature range. If the unit is to be used for supplying air for ventilation only, the convector must be sufficient in capacity to maintain a final air temperature of about 70 F. If the unit is to be used for heating as well as for ventilation, the convector must be sufficient to maintain the necessary final air temperature for the conditions involved.

3. *Control of Temperature.* This is accomplished by varying the temperature of the air discharged from the unit (1) by the automatic operation of a mixing damper which controls the relative quantities of air being blown through the heating unit or by-passed around it, (2) by operation of valves on different layers of convector surfaces, or (3) by variation in the temperature of the circulating heating medium.

The outside air inlet damper and recirculating damper (where one is provided) should be so connected that there will be an uninterrupted supply of air to the fans at all times the unit is in operation. These dampers may be operated by hand or by pneumatic or electric motors manually controlled from some central point.

These dampers may also be linked together, in the form of mixing dampers and be controlled by a thermostat in the cold air intake, by a differential thermostat acted upon by both the cold air and the recirculated air, or by a thermostat in the two streams of air after they are mixed, so as to keep the relative proportion of air taken in from out-of-doors commensurate with outside temperatures and to prevent drafts of cold air being blown through the unit into the room.

Provision should be made for the inlet damper to close automatically whenever the fans are shut down, and not to open until the room is properly heated when the fans are again started. The minimum temperature of the air delivered by the machine should be regulated automatically by a thermostat in the outlet air which controls the temperature of the heated convector, or this minimum temperature may be maintained by properly mixing the inside and outside air by means of the mixing dampers under thermostatic control referred to above. Another thermostat in the recirculated air intake to the unit or elsewhere in the room controls by-pass dampers or the supply of heating medium, or both, so as to control the temperature of the air leaving the unit according to the heat requirements of the room. In addition to these thermostats, a room thermostat is needed to control any other heat sources for the room. (See Chapter 14.)

Thermostats for controlling by-pass dampers must be of the intermediate type to hold the dampers in intermediate positions to prevent objectionable drafts. When direct radiators are used in conjunction with unit ventilators, the control is usually arranged so as automatically to open the valves to the direct radiators when the room temperature falls about 2 deg below the setting of the thermostat for the unit ventilator.

Another arrangement opens the radiator valve whenever the unit ventilator control reaches the full heating position. Further information on this subject is contained in Chapter 14.

4. *Distribution.* This function is governed by the proper selection and location of the unit. Diffusion and distribution are dependent upon a relatively high velocity air stream discharged in a generally vertical direction, and in order to insure satisfactory diffusion in the room the less the difference between the temperature of the air discharged from the unit and that of the room air, the better. With a final temperature above 110 F, excessive stratification of the air may be experienced. Troublesome drafts may be eliminated to a large extent if a static pressure is built up in the room.

5. *Recirculation of air* requires less fuel than does the use of all outside air and aids in heating up quickly. Certain units are designed to recirculate all air at all times, except when the admission of outside air is needed to regulate room temperatures. Under this arrangement, the outside air for ventilating purposes is obtained solely from infiltration, but the amount thus obtained may or may not be sufficient to meet legal ventilating requirements for public buildings. Recirculation of the air in schools is therefore prohibited by ordinance in many communities. Ventilating systems in schools should be arranged for taking in a sufficient quantity of air to constitute, with infiltration, not less than 10 cfm per occupant of a room.

6. *Quiet Operation.* Since the unit ventilator is generally set in close proximity to the room occupants, it must operate with exceeding quietness.

SPLIT AND COMBINED SYSTEMS

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is

heated. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be practically no heating.

LOCATION OF UNIT

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom.

VENTS

The size and location of the vent outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire, contamination, and other hazards.

CAPACITIES

Unit ventilators are available in air capacities ranging from 450 cfm to 5000 cfm and with corresponding heat capacities (above that required for ventilation purposes based upon an outside temperature of zero and an inside temperature of 70 F) ranging from 30 Mbh to 144 Mbh (1 Mbh = 1000 Btu per hour). Some manufacturers furnish a unit with several heating capacities for each air capacity, thus enabling the engineer to select the unit best adapted to the heating and ventilating load. Capacities should be determined in accordance with the A.S.H.V.E. Standard

Code for Testing and Rating Steam Unit Ventilators[3]. Typical capacities are given in Table 3.

The amount of heat to be supplied by the unit ventilator will depend on the amount of air passed through the unit and the temperature range through which the air is heated. The weight of air (W) to be circulated per hour is fixed by the ventilating requirements.

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulæ:

When all of the air handled by the unit is taken from the outside,

$$H_t = 0.24\,W\,(t_y - t_o) \tag{1}$$

$$W = dQ \tag{2}$$

$$t_y = \frac{H}{0.24W} + t \tag{3}$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, Btu per hour.
H_v = heat required to warm air for ventilation, Btu per hour.
H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per hour.
t = temperature to be maintained in the room.
t_o = outside temperature.
t_y = temperature of the air leaving the unit.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24\,dQ\,(t - t_o) \tag{4}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. $H = 24{,}000$; $d = 0.075$
$Q = 1000 \times 60 = 60{,}000$ cfh; $t = 70$ F; $t_o = 0$ F.

Substituting in Equation 4:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60{,}000\,(70-0) = 99{,}600 \text{ Btu}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60{,}000} + 70 = 92.2 \text{ F}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24W_o\,(t_y - t_o) + 0.24\,W_i\,(t_y - t) \tag{5}$$

[3]Adopted 1932. See A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.

$$W_o = d_o\, Q_o \tag{6}$$

$$W_i = d_i\, Q_i \tag{7}$$

where

d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cu ft per hr.
Q_i = volume of air taken in from the room, cu ft per hr.

$$t_y = \frac{H}{0.24\,(W_o + W_i)} + t \tag{8}$$

$$H_t = H + 0.24\, d_o\, Q_o\, (t - t_o) \tag{9}$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In Example 1, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirements H_t reduce from 99,600 Btu to 24,000 Btu, or to about one fourth. Such a unit handling one-third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu. Units designed and operated on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24\, W\, (t_y - t) \tag{10}$$

TABLE 3. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

CUBIC FEET OF AIR PER MINUTE	TOTAL CAPACITY IN SQUARE FEET OF EQUIVALENT DIRECT HEATING SURFACE (RADIATION)		CAPACITY AVAILABLE FOR HEATING THE ROOM IN SQUARE FEET OF EQUIVALENT DIRECT HEATING SURFACE (RADIATION)		FINAL AIR TEMPERATURE (DEG FAHR)
	EDR	Mbh	EDR	Mbh	
600	285	68	95	23	105
750	350	84	115	28	105
1000	455	110	150	36	105
1200	565	136	190	46	105
1500	705	169	235	56	105

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \tag{11}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

ATTIC FANS

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature of a house.

Because the low static pressures involved are usually less than ⅛ in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and they should be capable of giving about thirty air changes per hour. The two general types of attic fan installations in common use are:

Open attic fans, in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below. Fresh air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the central grille. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fan, in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Fresh air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

The locations of the fan, the outlet openings, and the grilles should be chosen after consideration of the room and attic arrangement in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

The operating routine which will secure best results with an attic fan is an important consideration. A typical routine might require that in the late afternoon when the outdoor temperature begins to fall, the windows

on the first floor and the grilles in the ceiling or the attic floor should be opened, and the second story windows should be kept closed. This will place the principal cooling effect in the living rooms. Shortly before bedtime, the first floor windows may be closed and those on the second floor opened, to transfer the cooling effect to the sleeping rooms. A time clock may shut off the fan before waking time, or the fan may be stopped manually at a later hour.

A disadvantage arising from the passing of a great amount of outside air through a house is the dust nuisance, which varies considerably in different locations. Persons suffering from allergic diseases caused by air-borne pollens will have their troubles increased with attic type coolers.

Some typical data on an attic fan installation in an average six-room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installation cost	\$75 to \$400, average \$250
Fan data	9000 cfm average, 280 rpm if belt driven, 570 rpm if direct connected, 500 watts input
Operating period	April 15 to October 15, intermittently as weather conditions demand
Power consumption	500 kwh per year for 8 months' operation

UNIT COOLERS

A unit cooler, as defined in Chapter 41, is a device usually comprising an extended-surface element and a motor-driven fan mounted integrally in a housing, suitable to be placed within or adjacent to the room served. The refrigerating medium is brought to the unit from an outside source, and the fan drives air over the cooling element; generally, no ducts are attached to inlet or outlet. With provision for filtering the air and taking in outdoor air for ventilation, the apparatus becomes a unit conditioner (Chapter 12). An alternative design uses chilled water or brine spray for cooling the air; it is essentially a small compact air washer with built-in fan and accessory equipment.

The principal field for unit coolers is in cold-storage plants, fur-storage vaults, packing houses, provision stores, brewery fermentation and stock rooms, and industrial process work. Coolers have, to a considerable extent, supplanted the bunker coils heretofore placed on ceilings and walls, because of demonstrated advantages with respect to: compactness, first cost, maintenance expense, damage from drips, ease of defrosting, maintenance of sanitary conditions, uniformity of temperature throughout the space served, and uniformity of temperature under variable load conditions, as well as control of humidity and circulation of room air when conducive to improved results.

A typical suspended unit is shown in Figs. 3 and 4. A motor-driven propeller-type fan is bracketed to the frame of a sheet-metal housing that contains an extended-surface coil, and a double set of louvers acting also as a moisture eliminator is provided at the outlet side. The horizontal louvers are adjustable to direct the air downward, horizontally, or upward, as desired. The lower part of the housing forms a drip pan, requiring a drain connection to dispose of the condensation when dehumidifying air

at usual room temperatures, or of the water when defrosting in low-temperature service. A cabinet-type unit for floor mounting is shown in Fig. 5; other designs are illustrated in the *Catalog Data Section* at the rear of this volume.

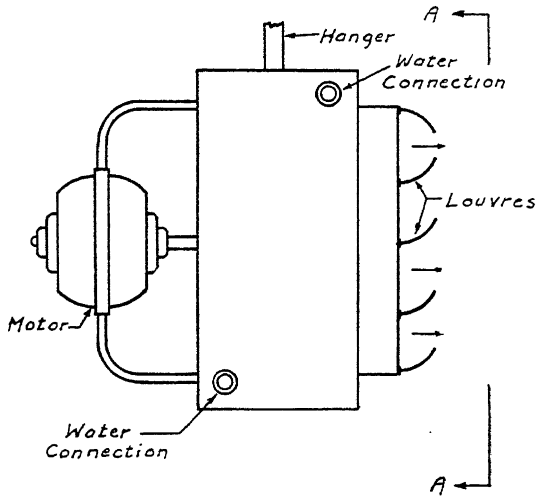


FIG. 3. CEILING UNIT

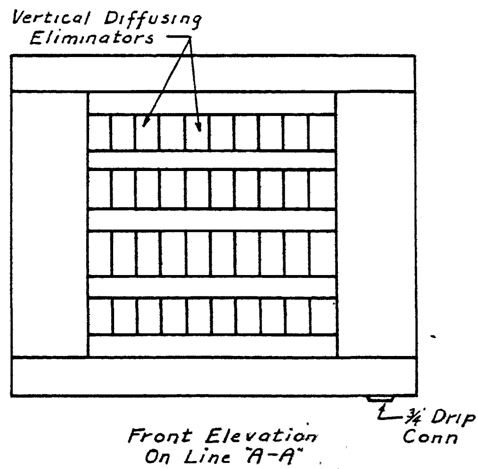


FIG. 4. ELEVATION THROUGH LINE AA

Depending upon the arrangement of the cooling coil, chilled water, brine, or a direct-expansion refrigerant may be employed. For cooling service at or near ordinary room temperatures, the considerations affecting a choice of cooling medium are those discussed in Chapter 12 for unit air conditioners. At lower temperatures, as for cold-storage, the

refrigerant system is usually dictated by the requirements of other refrigeration services supplied from the same condensing unit or from a central plant.

Details of construction employed in unit coolers are generally similar to those for unit air conditioners, with special attention paid to the use of non-corroding materials. Temperature control is obtained by starting and stopping the fan, with or without regulation of the cooling liquid or direct-expansion refrigerant admitted to the coil. Usually, a thermostatic control is provided ahead of the expansion valve at the inlet to the coil, tending to maintain constant temperature and pressure inside the coil regardless of cooling load, with a float at the outlet to prevent accumulation of liquid refrigerant in amounts sufficient to interfere with dis-

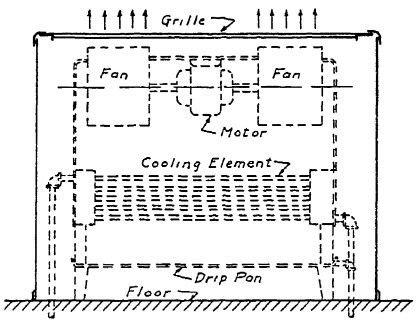


FIG. 5. CABINET TYPE COOLING UNIT

tribution between the various unit coolers served by a central condensing unit.

Ratings of unit coolers may be expressed in Btu per hour or in tons of refrigeration, with specified quantity, temperature, and humidity of air at the inlet, and with a stipulated pressure or temperature maintained within the cooling coil when using direct-expansion refrigerants. When chilled water or brine are used as the cooling media, the quantity and inlet temperature must be given. Ratings and dimensions of representative makes of unit coolers are given in the *Catalog Data Section.*

PROBLEMS IN PRACTICE

1 ● Is it better to use high pressure or low pressure steam in unit heaters?

The answer to this question depends upon the following circumstances: If steam is used only for heating purposes, it is usually best to design the entire system for low pressure steam. When steam is generated at high pressure for other purposes, it can be used either at full boiler pressure or at reduced pressure in the unit heaters. If the steam pressure is reduced, the heating elements should be capable of withstanding the full boiler pressure. When steam at full boiler pressure is used in the heating elements, the heating surface should be reduced so that the outlet temperature will not be more than 70 F higher than the inlet temperature. With the use of high pressure steam special care must be exercised in venting the units of air, in preventing flash steam in the returns, and in preventing corrosion from superheated returns.

2 ● How should heat losses be calculated for a building using unit heaters?

The heat losses should be calculated in exactly the same manner as for any other heating system. If the method of calculation takes into consideration the variation in temperature from the floor to the ceiling, the temperature variation should be reduced when calculating the heat losses for a unit heater job. This is advisable because with unit heaters the temperature variation between the floor and the ceiling is from ½ to 1 F per foot of elevation, whereas with direct radiators or pipe coils, this variation may be twice as great. Unless the ceiling height is more than 15 ft, the temperature variation between the floor and the ceiling is usually neglected when unit heaters are used.

3 ● On what basis should unit heaters be selected?

Unit heaters should be selected to furnish enough heat to offset the heat losses and to circulate the air in the room fast enough to provide good heat distribution. In the average building, if the outlet temperature does not exceed the inlet temperature by more than 70 F, sufficient air capacity will usually be provided for proper circulation if the units are selected strictly on the basis of heating capacity. However, if the units are hung unusually high or if the heat loss is low in proportion to the volume of the room, then, in order to obtain the desired air capacity, it is usually necessary to employ more heaters than are required to offset the normal heat loss. Inasmuch as the heat distribution depends upon the outlet temperature, the outlet velocity, the character of air flow from the heater, the height at which the heaters are hung, and the size of the heater itself, the manufacturers' literature should be carefully studied in determining the exact number of heaters to be employed.

4 ● Is it satisfactory to use superheated steam in unit heaters?

Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on the saturated steam temperature and not on the total temperature. If unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

5 ● Is it satisfactory to install one unit heater as the total load on a coal fired boiler?

Such an arrangement is impractical if the unit heater is started and stopped in keeping with the room temperature. However, if the room temperature controls the steam pressure and the unit heater is arranged to start when there is steam in the mains and to stop when there is no steam in the mains, such an installation will be satisfactory.

6 ● Will a unit heater with a slow speed fan be more quiet than one with a high speed fan?

The speed of the fan is no indication of quietness. Quietness is a function of the type, diameter, blade form, speed, and location of the fan.

7 ● Is it satisfactory to use steam at pressures less than atmospheric for unit heaters?

If the air inlet temperature is above freezing, steam at any pressure may be used in the unit heater. If the inlet temperature is below freezing, steam of at least 5 lb pressure (or with a positive 5 lb pressure differential between supply and return) should be used, and the steam supply should never be throttled or the heating element may be frozen.

8 ● In general, what is the primary function of a unit ventilator?

To maintain the desired room air conditions as to temperature, air change, and air cleanliness, without drafts regardless of variations in outdoor temperature, occupancy, sun heat, and wind.

9 ● What are the usual working parts of a unit ventilator?

A fan and motor assembly, a set of heating elements, outdoor and indoor air dampers, filters, outlet grille, some method of controlling the outlet temperature above a minimum of 60 F, and some method of varying the outlet temperature in keeping with the room requirements. All of these parts are usually enclosed in an attractive steel cabinet in which the piping is concealed.

10 ● Do all unit ventilators introduce a constant amount of outdoor air?

Certain types employ full recirculation except when outdoor air is obtained by throttling the steam valve on the heating element so the proportion of outdoor air to room air is varied. This is a very economical type of unit ventilator but in some communities it cannot be used because of existing laws which require that some fixed amount of outdoor air be introduced whenever the room is occupied. Certain types of units are designed to always take in a minimum quantity of air from the outside and to automatically vary this with the weather.

11 ● Where should a unit ventilator be located?

In the center of the longest outside wall under the windows.

12 ● What further precaution should be taken in locating unit ventilators?

With most unit ventilators, a high velocity jet of air is discharged toward the ceiling at a slight pitch toward the room; all unit ventilators should be installed in such manner that this jet is not interfered with. For this reason the air should be distributed on a flat ceiling without beams, but if beams are present, the unit ventilator should be so located that the air will be discharged parallel to the beams.

13 ● When unit ventilators are installed to employ variable recirculation, what special precautions are necessary?

Where partial recirculation is employed, some effective means should be installed within the cabinet of the unit ventilator to prevent unheated outdoor air from being blown into the room through the room air opening while the unit is mixing indoor and outdoor air. This means may be self-operating dampers placed in the path of the room air, or filters so arranged that the outdoor air must pass through them before it can enter the room.

14 ● Generally speaking, should direct radiators be used in addition to unit ventilators in school classrooms?

The best practice in schoolrooms is to place as much heating capacity as possible in the unit ventilator itself. However, in selecting the unit ventilator, the outlet temperature should not exceed 110 F and the rate of air circulation should not exceed 9 room volumes per hour (anemometer measurement) or 7½ room volumes per hour (A.S.H.V.E. Code measurement). If the heating capacity under these conditions is sufficient to heat the room, no additional radiation is required. If the heating capacity is not sufficient, direct radiation should be used to make up the required total. Radiators always tend to offset the chilling effect of cold walls and windows quicker than warm air does.

15 ● Are vent outlets required with unit ventilators?

Though experience has indicated that in practically all school and office buildings the cracks around the windows, doors, and baseboards are so numerous that vents are not required, in many communities vents are required by existing laws. In some cases the sizes are also stipulated in the laws. When the size is not stipulated, vents should be designed on the basis of a velocity not greater than 600 ft per minute. Vent flues should always be provided with a damper in order that they may be throttled.

Chapter 14

AUTOMATIC CONTROL

Apparatus Sensitive to Temperature, Apparatus Sensitive to Relative Humidity, Apparatus Sensitive to Pressure, Accessory Apparatus, Temperature Control Systems, Control of Automatic Fuel Appliances, Individual Room Control, Zone Control, Industrial Processes, Air Conditioning Systems, Seasonal Operation

AUTOMATIC controls can be installed on any type of heating, ventilating, or air conditioning system to maintain desired conditions automatically, and with maximum operating economy. The variety of automatic control equipment available is such that a suitable control system can be devised without difficulty, provided that the conditions to be maintained are known and the control equipment is properly chosen. This chapter outlines briefly the various types of control apparatus and indicates the method of their application to typical heating, ventilating, and air conditioning systems. Specific control devices and systems are described in the *Catalog Data Section* of THE GUIDE.

Controls are applied for the following reasons:

1. To maintain conditions required for human comfort and efficiency.
2. To maintain conditions required for industrial processes.
3. To obtain economy in operation.
4. To provide necessary safety measures.

CONTROL APPARATUS

The various pieces of control apparatus may be grouped under the following general headings:

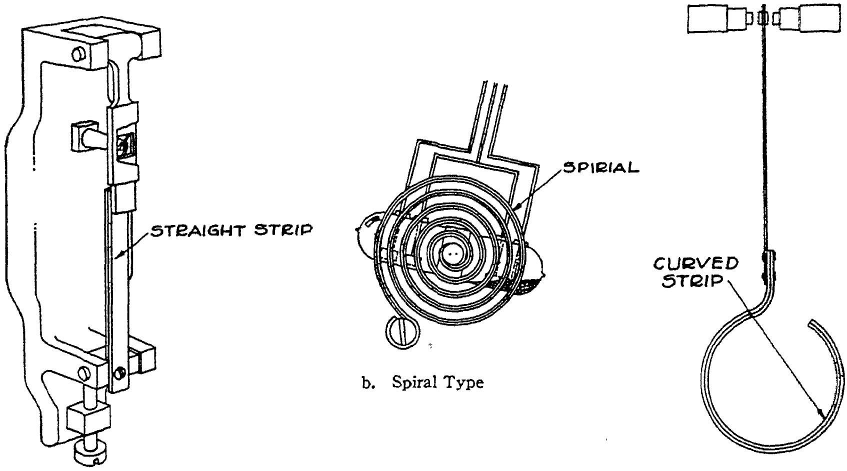
Apparatus Sensitive to Temperature

Temperature-sensitive devices which will respond to changes in temperature, and which will motivate equipment to compensate for the changes, are usually called *thermostats*. They have many specialized forms for use in specific control applications. Thermostats are the detectors of a control system which identify changes in desired temperature conditions and automatically call for compensating action.

Thermostats are actuated by various means, all of which have the common characteristic of responsiveness to small changes of temperature. The actuating element may be a piece of bi-metal in straight, helical, or spiral form (Fig. 1), which, by bending slightly as the temperature changes, actuates an electric or pneumatic *switch* to govern the controlled apparatus; or the actuator may be a diaphragm, bellows, or tube filled

with a volatile liquid (Fig. 2) in such way that expansion and contraction with changes in temperature will operate the controlled apparatus by a direct mechanical, electric, or pneumatic connection.

A *room* or *wall thermostat* in its simplest form contains a single temperature-sensitive element which is so set that it maintains, by actuating the controlled system, a single temperature. The *two-temperature* or *dual thermostat* has two temperature-sensitive elements, one of which is set for a higher temperature than the other. Such a thermostat is used on day-night systems where the night temperature is to be lower than that



a. Straight Strip Type

b. Spiral Type

c. Curved Strip Type

FIG. 1. TYPICAL BI-METALLIC THERMOSTATIC ELEMENTS

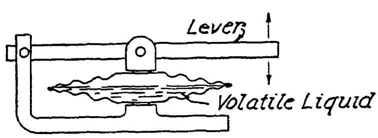


FIG. 2. DIAPHRAGM TYPE THERMOSTAT

maintained during the daytime hours. Switching the control from one element to the other is accomplished by an external or an internal switch, which can be operated manually or by a time device.

Duct type thermostats are used in systems where the equipment must respond to changes in the temperature of the air passing through a duct. In their usual form, these thermostats are so constructed that their switching mechanism is outside the duct, while the temperature-sensitive element projects inside into the air stream.

Thermostats which operate in liquids have the same general construction as duct thermostats except that the sensitive element is usually enclosed in a tube to keep it from direct contact with the liquid. They

are used in pipes, vats, and tanks, and are called *immersion thermostats.* Such a thermostat is illustrated in Fig. 3.

Sometimes *surface thermostats* are used in place of duct or immersion thermostats. These devices, so constructed as to respond to changes in temperature of the surface of the duct or vessel containing a fluid, are clamped or screwed to such surfaces in a manner which will provide as rapid as possible heat transfer between the surface and the sensitive element.

Apparatus Sensitive to Relative Humidity

Devices which are responsive to changes in the relative humidity of the surrounding air, and which will motivate equipment to compensate for the changes, are called *humidistats* or *hygrostats.* These may vary considerably in their sensitive elements, but they all operate through connecting equipment which automatically causes humidifying apparatus to supply more or less moisture as required. Some of the more complicated

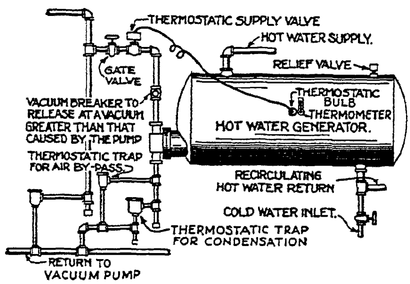


FIG. 3. SELF-CONTAINED THERMOSTAT ON HOT WATER TANK WITH VACUUM RETURN

ones contain essentially two thermostats, one working on a dry-bulb temperature and the other on a wet-bulb temperature; by proper interconnection of the parts they operate to maintain a definite relation between these two temperatures. Other devices use elements, directly sensitive to humidity, made of special wooden blocks, human hair, fiber, membranes, or strips of prepared paper. Hygrostats are available for use with both electric and pneumatic control systems.

Apparatus Sensitive to Pressure

Use is made of devices which are responsive to changes in pressure, and which will motivate equipment to compensate for the changes. Such devices usually depend upon the flexing of a diaphragm or bellows as caused by varying pressures or vacuums to obtain the mechanical movement necessary to actuate an electrical or pneumatic switch.

Apparatus Which Operates Valves

Apparatus which is so mechanically or electrically equipped that it will open and close valves, and possibly give them fixed intermediate positions in any pipe line of a heating, ventilating, or air conditioning system, is termed a *valve operator.* The function of a valve operator is, essentially,

to move the plunger of a valve in a manner required by its type and construction. For instance, in a single-seat valve, the disc is moved against the seat and held there with sufficient pressure to prevent flow. A three-way valve, however, requires a valve operator that will place the double disc, as required, between the two seats. Each type of valve has special characteristics to which a valve operator must be adapted.

When a valve is used in shut-off service the valve operator simply opens the valve or closes it completely, as required. When the valve is to provide throttling service, a different type of valve operator is used so that the valve may be held at any intermediate position between open and closed. Valve operators use as their power source either compressed air (pneumatic system), electricity (motor-driven type of solenoid type), or a volatile liquid (direct-connected type).

Apparatus Which Operates Dampers

Apparatus which is so mechanically or electrically equipped as to open and close dampers, and possibly give them fixed positions, in accordance with the purposes of the system using the dampers is termed a *damper operator*. Damper operators are made for opening, closing, and positioning the dampers in the ducts of heating, ventilating, or air conditioning systems in the same way that valve operators regulate the valves. They receive their signals from thermostatic or manual switches.

The sources of power used are compressed air, electricity, or volatile liquids. The damper operator is connected to its damper by direct connection or by a linkage, according to conditions, and it can usually be mounted either outside or inside the duct in which the damper is located.

Accessory Apparatus

Accessory apparatus is that additional equipment at the terminals of a control system necessary to make it operative. Every temperature control system requires a number of accessories, which will vary with the different types of systems. For instance, pneumatic systems require a compressor and a storage tank for the air which operates the units, and low-voltage electric systems require a transformer or generator to provide the required current.

Most of the larger control systems will have some sort of central switchboard which may include indicating and recording devices as well as control switches. Thermostat guards are generally used in gymnasiums, schools, and places of assemblage for protective purposes. Time switches and similar devices are often important parts of certain types of control systems. Couplings, mountings, and indicators are often parts of a system.

Connecting Apparatus

Connecting apparatus is that equipment used to connect the various parts of a control system. Because the parts of the system are often some distance apart, the connecting means are important, and the connections must be properly planned and made.

The connecting elements are fairly obvious. The pneumatic system uses compressed air carried in small pipes and tubing. Electric systems

are wired for low-voltage or high-voltage power supply. Systems employing volatile liquids generally use flexible tubing if there is distance between the sensitive bulb and the operating unit. Each form has certain limitations which the designer of the system must consider.

Since few control installations are alike, the manufacturers of control apparatus usually maintain engineering departments staffed by experienced men whose advice may be had on control problems. Progress in automatic control has been rapid in the past few years and the field of automatic control has become specialized.

TEMPERATURE CONTROL SYSTEMS

The control of direct radiation is simple. Each radiator has a valve on its steam or water supply, with a thermostat to govern the opening and closing of the valve to maintain the desired uniform temperature. One thermostat may control the valves on all the radiators in a room, or, if the room is large, more than one thermostat may be used, with each one governing one radiator or a group of them. Unit type thermostatic valves may be used, one on each radiator.

The location of wall thermostats is important. They must be on inside walls where they will not be affected by drafts of either warm or cold air, but where they will be exposed to general room conditions. If vibration is present, they must be mounted on shock-absorbing bases. If the walls are abnormally hot or cold, the thermostats must be mounted on heat-insulating bases. The connecting means can be concealed in the wall, under the floor or ceiling, or behind baseboards or moldings.

Modulating type valves cannot be used successfully on one-pipe steam systems because the partial opening of valves will not allow the condensate to escape against the incoming steam.

A discussion of steam heating systems is given in Chapter 31, and further information on control requirements of direct radiation may be obtained therefrom.

Control of Unit Heaters

Unit heaters are commonly ceiling-hung or floor-mounted units consisting of a steam or hot water coil with a fan behind it to force air past the coil and into the room. Vanes direct the warm air flow. The simplest and commonest way to control a unit heater is to have in the heated space a thermostat which will turn on the fan when heat is required and shut it off when the demand is satisfied. However, where there is natural circulation through the unit, it is advisable to put a valve on the steam or hot water supply line and arrange it so the steam will be turned on only when the fan is running.

As a precaution against allowing the unit heater motors to continue to run if the steam supply fails or is for some reason shut off, either a pressurestat or a thermostat in the supply line, or a thermostat on the return line may be installed to stop the motor when the pressure or temperature in the supply line, or the temperature in the return line, drops below a predetermined point. When the fan and the steam are controlled simultaneously, such thermostat will also prevent the blowing of cold drafts.

The net result in any case will be that the fan will run only when there is heat in the coil.

Control of Unit Ventilators

The unit ventilator presents a different control problem than the unit heater. Generally this type of unit draws its supply of air from the outside, heats it, and introduces this air into the room under control. There are many types of unit ventilators on the market. Some have a mixing damper by which the temperature of the air entering the room may be varied, others have valves for this purpose, and still others use a combination of the two. Regardless of the construction of the machine, the essential requirement is that the temperature of the air delivered to the room should change slowly and remain as near room temperature as possible. Frequently direct radiators are used in conjunction with the unit ventilators to supply additional heat in extremely cold weather or for quickly heating up the room.

The four general types of control for unit ventilators are as follows:

1. A damper operator, which is controlled by a room thermostat, is attached to the mixing damper. When the thermostat calls for heat, the damper is moved to a position which forces more air through the heating unit and thus increases the amount of heat supplied to the room. This action must be gradual so that the air temperature may be changed slowly to prevent the drafty condition caused by supplying first hot and then cold air. This simplest arrangement is often condemned because it frequently results in drafts.

2. In mild weather the heating unit frequently supplies sufficient heat to cause overheating of the room, even though all of the air is by-passed around the heating unit. To avoid this fault a valve is placed on the heating unit to close the steam supply when the damper is by-passing all of the air. This valve is used in addition to the damper operator explained in the foregoing paragraph, but though giving better results, it may fail to prevent drafts.

3. In some unit ventilators one or more heating units are used without a mixing damper. A gradual-acting valve on each heating unit controls the supply of steam to the unit to give the proper amount of heat required to maintain the desired room temperature. A thermostat to govern each valve may be installed in the room, or one thermostat may be used for all valves, but unless a thermostat is placed directly in the air stream of each unit, drafts may be encountered.

4. Another type of unit ventilator is arranged so that all recirculated air passes through the heating unit, and the outside air is introduced into the room for cooling purposes only. The outside air damper and the recirculated air damper are interlocked so that one damper operator will control them. In addition a valve operator is placed on the heating unit. Both of the operators should move gradually to avoid drafty conditions. When the thermostat calls for heat, the damper operator slowly closes the outside air damper and simultaneously opens the recirculating damper; if this does not meet the demand, the valve on the heating unit opens until the room temperature reaches the desired point.

For additional information on the control of unit ventilators, refer to Chapter 13.

Central Fan Heating and Ventilating Systems

The numerous types of central fan systems present many control problems. In general they all have one point in common, namely, that the temperature change may be very fast because of rapid circulation.

System for Ventilating Only (Split System). Fig. 4 shows an accepted control for ventilating systems. Thermostat *A* located in the outside air

duct is set just above freezing, and controls a valve *C* on the first heating coil. This valve is either completely open or completely closed. The by-pass damper *B* and the other two valves *D* and *E* are controlled by a duct thermostat *F* located in the discharge duct from the fan. If the temperature of the air surrounding the thermostat *F* increases, the damper is moved automatically to admit more cold air. Should this not reduce the temperature sufficiently, the valves on the heating coil will be closed gradually and in sequence until the correct temperature is reached. The opening or closing of the damper *B* and the valves *D* and *E* must be gradual or there will be a wide fluctuation in air temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the room. Hence, in order to maintain

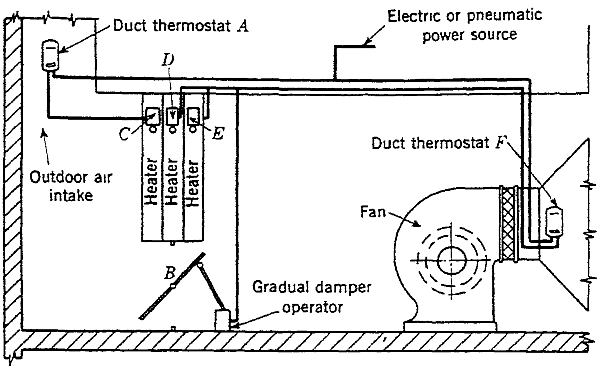


FIG. 4. CONTROL OF A SPLIT SYSTEM OF VENTILATION

controlled room temperatures it is necessary to use room thermostats governing control valves placed on the radiators. With this type of central fan system it is possible to ventilate a large number of rooms by means of one fan.

In some installations, such as in theaters or auditoriums, it is difficult to install sufficient direct heating surface to offset the heat losses from the room. Also there are installations where a short heating-up period is allowed before occupancy of the room, and it is advisable to use the entire heating capacity of the ventilating system for this purpose.

In central fan systems, air washers are often used and in such cases, due to the effect of temperatures on humidity, additional control is required. Fig. 5 shows such an arrangement with control of the second tempering heating unit by the air washer temperature and with the usual control of the first tempering heating unit by the outside temperature. This permits the air to be kept cool while passing through the washer so that too much moisture will not be absorbed. Fig. 5 also shows control of the reheating units by a duct thermostat in the fan discharge, and the application of a pilot thermostat to a system of this sort.

Combined Systems. There are various central fan systems which are used for both heating and ventilating. They are usually arranged with tempering heating units, automatically controlled to provide a minimum temperature for ventilating only, and additional heating units to supply the heating requirements. Fig. 6 shows a type of system which has the reheating units located in the fan room. Tempered air at about 70 F is supplied to the fan. It may be further heated by the reheating units, or it may pass into the tempered air chamber. A room thermostat controls a gradual-acting damper operator on the double mixing damper in the warm and tempered air chambers. When the thermostat calls for heat, the

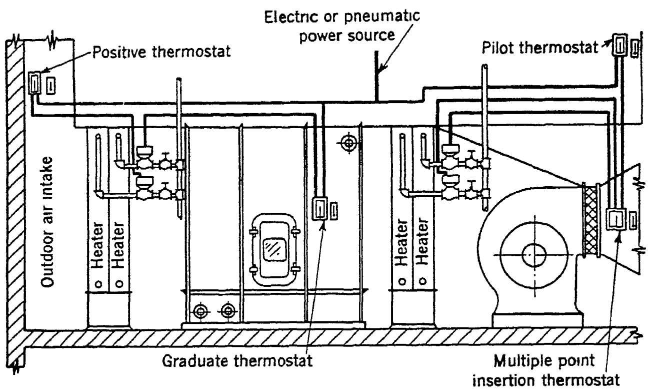


FIG. 5. USE OF PILOT THERMOSTAT ON VENTILATING SYSTEM WITH AIR WASHER

damper operator moves the dampers so that more air is taken from the warm air chamber. It is essential that the double mixing damper be moved slowly to prevent alternate blasts of hot and cold air from being supplied to the room.

Outside Air, Recirculating, and Vent Dampers. In all types of plenum systems, the outside air damper is usually opened and closed by a damper operator. This operator may be controlled from a switch in the engineer's room or it may be operated by a relay in the fan motor circuit. When the ventilating fan is started, the relay causes the damper operator to open the outside air damper.

Recirculating dampers and vent dampers may also be opened and closed by means of damper operators controlled from remote locations. Generally these damper operators are positive acting and are either completely opened or closed. However, in some cases where part outside air and part recirculated air is used, it is advantageous to use damper operators which have a certain number of definite positions. With this type of operator it would be possible to use 75 per cent outside air and 25 per cent recirculated air, or any other proportions which might be predetermined. These damper operators are controlled from switches generally mechanically interlocked so that the total opening of the two dampers is 100 per cent.

Hand-Fired Coal Systems

In small buildings the heating plant may be controlled by a single thermostat located in a key room in the building, instead of each room having its own control.

The most common control for a hand-fired furnace or boiler consists of a room thermostat and a furnace regulator of some type. The thermostat should be located in a representative room; never, of course, near the chimney or heat flue, too close to a radiator, or in a drafty hallway, and preferably on an inside wall. The regulator is attached to the draft and check dampers of the furnace. When the temperature of the air sur-

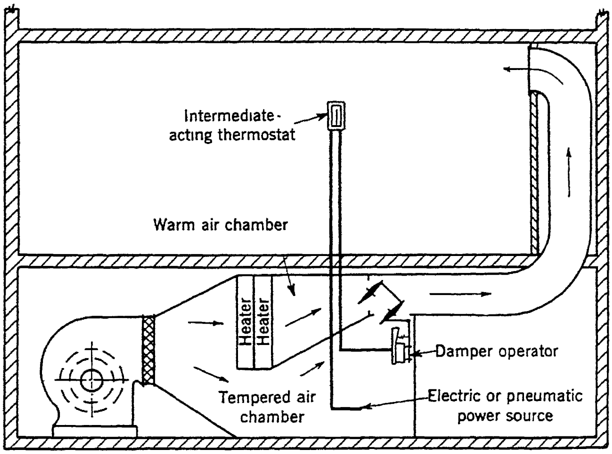


FIG. 6. CONTROL OF MIXING DAMPERS WITH INTERMEDIATE-ACTING THERMOSTAT

rounding the thermostat drops, the thermostat causes the furnace regulator to open the draft and close the check damper. As soon as the room comes up to temperature, the draft is closed and the check damper opened. With this arrangement on hot water heating systems it is advisable to install an immersion thermostat in the boiler. This thermostat should be connected with the room thermostat so that both must call for heat before the draft is opened, but either one may cause the draft to be closed. On warm air systems it is advisable to use a bonnet thermostat and on steam heating systems a pressure limiting device, in series in each case with the room thermostat. If the temperature of the heating medium becomes too high, the drafts will be closed even though the room thermostat continues to call for heat.

There have been some recent improvements in controls of this type, involving the use of special types of thermostats and auxiliary apparatus which will give closer control and prevent overheating in mild weather.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic temperature control be used with oil burners, gas burners, and stokers to aid economical operation. There are

many types of burners and many types of control, but there are some points common to all. First, a room thermostat is located in a key position in the building to maintain a given temperature at that point. Safety devices are installed in connection with this thermostat so that a failure of the ignition, power, or fuel supply will shut the system down. The same limit controls as recommended for coal burning should be used.

Oil Burners

Fig. 7 illustrates diagrammatically the essentials of an oil burner control circuit. Three thermostats are employed as shown in the illustration. Thermostat No. 1 will stop the burner when the room temperature is too high and No. 2 will stop the burner when the temperature of the heating medium exceeds the setting of thermostat No. 2. Both temperatures must be below their respective thermostat settings to start the burner. Thermostat No. 3 responds to the flame temperatures and in conjunction with the control switch acts as a safety to stop the burner if the latter fails to ignite or burn properly as demanded by thermostats No. 1 and 2.

Domestic Applications

Steam and hot water heating plants are often used to provide heat for the domestic hot water supply as well as for heating the building. Fig 8 illustrates one such system. The burner control is similar to that shown in Fig. 7 except that either the room thermostat or the tank thermostat may start the burner. If the house is warm enough, the house temperature control valve will remain closed, and the boiler, through the coil heater, will warm the water in the storage tank when the tank thermostat starts the burner. The burner will stop only when both thermostats are satisfied, or when the steam pressure shall have reached that allowed by the pressurestat. Much the same control is applied to gas burners and automatic coal stokers.

Gas Heating Appliances

On account of the ease and effectiveness with which the fuel can be controlled, gas-burning appliances are particularly adaptable to full automatic control. Standard equipment on a steam boiler generally includes provision for control through a room temperature thermostat, a steam pressure regulator, and a device which shuts off the gas in the event that the water level becomes too low. Practically all gas boilers are or may be equipped with automatic safety pilots which shut off the gas if the pilot flame is too low.

Water boilers are adapted to operation under thermostatic room temperature control and are also provided with water temperature control equipment. Warm air furnaces can be under the control of thermostats in the spaces being heated, as well as thermostats located in the heat ducts for the purpose of preventing unpleasantly hot air reaching the heated spaces. Variations in the pressure under which the gas is supplied to the appliance are controlled by means of a gas-pressure regulator. This is an essential part of practically all makes of gas-burning heating appliances; in fact, a gas-pressure regulator is required by the *American Gas Association* on all approved gas boilers, warm air furnaces (except floor furnaces), and unit heaters.

INDIVIDUAL ROOM CONTROL

The most elaborate type of automatic control is that by which the temperature in each room or in a group of rooms can be controlled. A thermostat in each room governs the valves on the radiators in that room,

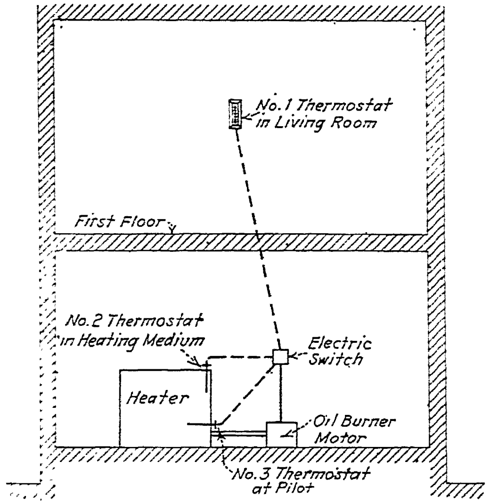


FIG. 7. ELECTRIC THERMOSTAT APPLIED TO OIL-FIRED HEATING SYSTEM

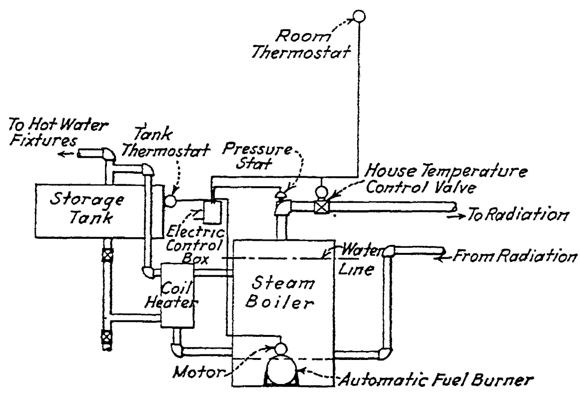


FIG. 8. TYPICAL ARRANGEMENT OF STEAM OR VAPOR SYSTEM WITH TWO THERMOSTATS CONTROLLING AUTOMATIC FUEL BURNER USED FOR HOUSE HEATING AND WATER HEATING

opening them as heat is called for and shutting them when the room is warm enough. The thermostats are all connected in relay so when any thermostat is calling for heat, an automatic burner will supply steam, hot water, or warm air, to the system; and when all the thermostats are satisfied, the burner will shut off. This is an excellent arrangement for larger

residences, and it may be applied, in modified form, in houses which have one room or a section that is difficult to heat.

ZONE CONTROL

Zone control is a step between a single thermostat and individual room temperature control. The building is first divided into sections or zones which may have quite different heat requirements. With this method of control:

First: The zoning should be done with reference to the compass, since the north and west quarters in most localities require considerably more heat during the heating season than do the south and east quarters.

Second: Most large office buildings have more or less space occupied by merchants, and some by clubs, or restaurants, which have short hours of occupancy. Much can be accomplished in zoning with reference to the kind of occupancy of space. For additional information on this subject, refer to Chapter 31.

Variations of the usual zone control methods by the use of recently developed special devices have been quite successful in obtaining greater economy from heating systems. Frequently these use an outside thermostat or group of thermostats which adjust the operation of the controls to conform to variations in weather conditions.

COOLING UNITS

Cooling units are readily adaptable to thermostatic control. Several arrangements are as follows:

1. Room thermostat in conjunction with a magnetic or motor-operated valve to regulate the flow of refrigerant to coil. Usually the fans operate continuously.
2. Room thermostat to control the operation of the compressor. The fans operate continuously.
3. Room thermostat to control the operation of the fan motors.
4. Room thermostat to control the operation of the fan motor and the compressor motor simultaneously.
5. Room thermostat to control the operation of the compressor with back pressure control to regulate the fans.

INDUSTRIAL PROCESSES

There are many industrial processes requiring automatic temperature and humidity regulation. The control equipment operates on the same principles that have been described, but it is often especially designed for each particular process. Each installation, or the installation for each process, is likely to be a problem peculiar to that process.

AIR CONDITIONING SYSTEMS

The following fundamental principles should be borne in mind in the solution of problems involving the control of air conditioning systems:

1. Dew-point temperatures vary only with the amount of moisture. That is, no matter how much a given mixture of air and water vapor is heated or cooled, the dew-point temperature remains the same, as long as there is no addition or subtraction of water. Cooling below the dew-point temperature will, of course, cause condensation of the water vapor. Also, at the same temperature, there is always the same proportion of water vapor in the saturated mixture, provided sufficient water and time are furnished for saturation.

Table 5, Chapter 1, shows the amount of moisture required to saturate a space at various temperatures. When the proper amount of moisture is determined, it is only necessary to set the air washer (dew-point) thermostat for the corresponding temperature of saturation; then if the air entering the washer has more humidity than desired, the excess will be condensed; and if it has less, the deficiency will be absorbed from the sprays.

For example, the dew-point temperature at 70 F and 40 per cent relative humidity is 45 F. Therefore, if the air temperature is maintained at 45 F as it leaves an air washer (assuming it is fully saturated) and then is heated to 70 F, it will have a relative humidity of 40 per cent. If it is desired to maintain these conditions in a given space, the air temperature can be raised to any necessary point, say 120 F (at which the relative humidity will be only 9 per cent). When the heat in the air has been dissipated, the space temperature being maintained at 70 F, the relative humidity will be 40 per cent.

2. Within ordinary operating ranges, saturated air will have a relative humidity of approximately 50 per cent when its temperature is raised 20 deg. For example, saturated air at 40 F raised to 60 F has a relative humidity of 48 per cent; 60 F saturated air raised to 80 F has a relative humidity of 50 per cent. (See Table 4, Chapter 1.) Thus a differential thermostat can be used to maintain a nearly constant relative humidity of 50 per cent by holding the dew-point temperature 20 deg below the dry-bulb temperature.

3. The *total heat* of the air and the water vapor mixed with it varies directly with the wet-bulb temperature. For example, the occupants of an auditorium give off sensible heat which tends to raise both the dry-bulb and the wet-bulb temperatures of the space; but the occupants also give off moisture which increases the absolute humidity and tends to further raise the wet-bulb temperature by an amount which is a direct indication of the heat expended by each occupant in evaporating this water. This relationship is useful in regulating the total heat, as wet-bulb temperatures can be controlled directly by means of a thermostat having a sensitive element covered with water-fed wicking, similar to a wet-bulb thermometer.

For example, the total heat of air at 80 F and 60 per cent relative humidity is the same as for air saturated at 70 F, *i.e.*, 33.5 Btu per pound, both having a wet-bulb temperature of 70 F. Air at 80 F and 60 per cent relative humidity (70 F wet-bulb = 33.5 Btu per pound) reduced to 70 F and 50 per cent relative humidity (58½ F wet-bulb = 25.2 Btu per pound, total heat) must give up 8.3 Btu per pound. If the sensible heat and moisture pick-up in an auditorium is 8.3 Btu per pound of air handled in the conditioning system, the wet-bulb temperature of the air entering the space must be maintained at 58½ F to secure a final condition of 80 F and 60 per cent relative humidity.

Control of Relative Humidity

The following are the most commonly used methods of controlling relative humidity:

1. A thermostat is located in or at the outlet of a spray-type air conditioner which maintains a constant saturation temperature of the air leaving the conditioner by varying the temperature of water entering the suction of the pump supplying the spray nozzles, or by varying the temperature of the air entering the conditioner, or both. The temperature of the air entering the conditioner may be varied by use of tempering heaters, or by the proper proportioning of supply and return air entering the conditioner. This thermostat is known as a dew-point thermostat, as it determines the dew-point temperature of the air introduced into the conditioned spaces. A second thermostat in the room, or in the path of the air leaving the room, maintains a constant dry-bulb temperature by varying the amount of sensible heat added to the air leaving the conditioner, or by varying the volume of air introduced into the conditioned spaces. These two thermostats, in combination, control the dry-bulb and dew-point temperatures, which accordingly fix the relative humidity.

2. A wet-bulb thermostat is located in the room, or in the path of the air leaving the room, to maintain a constant wet-bulb temperature by varying the saturation tempera-

ture at the air conditioner outlet. A dry-bulb thermostat is located in the room to maintain a constant dry-bulb temperature, which in combination with a constant wet-bulb temperature fixes the relative humidity.

3. A differential thermostat may be used to control relative humidity. This instrument consists of two thermostatic elements, one of which is in the path of the air leaving the conditioner, and the other under the influence of the dry-bulb temperature in the room. Instruments of this kind maintain a constant relative humidity by maintaining a constant difference between the dew-point temperature and the dry-bulb temperature in the room. (See Item 2 under Air Conditioning Systems.) One thermostatic element may be equipped with a moistening device to permit it to operate on wet-bulb temperatures. Such an instrument can be used to control the wet-bulb depression and thus the relative humidity.

4. A humidistat which responds directly to changes in humidity may be used to maintain a predetermined relative humidity with constant or with varying temperature. It may do this by varying the dew-point temperature of air leaving a conditioner; by varying, with dampers, the proportion of moist and dry air; by varying the amount of moisture otherwise added to the air; or by varying the dry-bulb temperature.

Humidification for Residences

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin No. 48*[1], as follows:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.
2. An effective temperature of 65 deg[2] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69.5 F with relative humidity of 40 per cent is the most practical for the attainment of 65-deg effective temperature.
3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.
4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.
5. The problems of humidity requirements and limitations cannot be separated from considerations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.
2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

Central Fan Air Conditioning Systems

In central fan air conditioning systems as described in Chapters 9 and 22, varying amounts of outside and recirculated air are used, except where contamination prevents re-use, and in general for obtaining humidity control under winter conditions heat is supplied to the air after it has passed the air washer. There are many control variations in use, and

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[1]See Humidification for Residences, by A. P. Kratz (*University of Illinois, Bulletin No.* 48).

[2]Sixty-six deg is the optimum winter effective temperature recommended by the A.S.H.V.E. Committee on Ventilation Standards. See Chapter 2.

Fig. 9 shows a composite diagram, rather than a system of control for a single installation. The control valves for a dehumidifying air washer are shown in Fig. 10. The functions of the control devices shown in Figs. 9 and 10 are as follows:

Winter Operation (With Steam)

1. Thermostat *A* opens a direct-acting valve in the steam supply to a low-capacity tempering coil *P*. The thermostat is set at 35 F.

2. Thermostat *B* in the path of the air leaving the second tempering coil *Q* controls a valve in the steam supply to the coil *Q* at 45 F.

3. Thermostat *C* controls the intake *M* and return air *N* dampers at 50 F. This location of thermostat *C* is primarily for operation with steam heating and at such times as by-pass damper *O* is closed. See discussion under the heading *Spring and Fall Operation*.

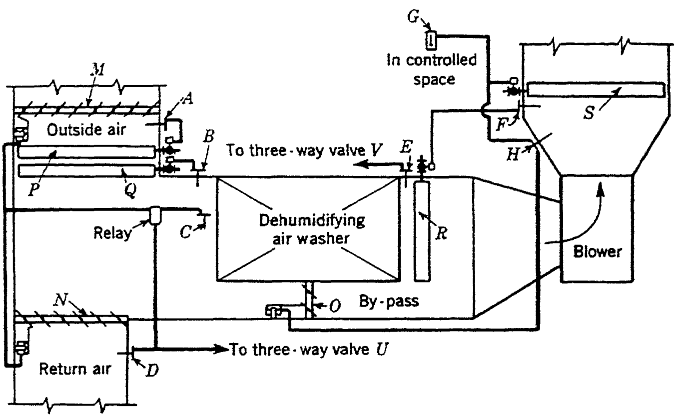

FIG. 9. DIAGRAMMATIC ARRANGEMENT OF VARIOUS PHASES OF CONTROL FOR A CENTRAL FAN AIR CONDITIONING SYSTEM

4. Humidistat or wet-bulb thermostat *D* in the return air, acting through a relay, causes *C* to call for outside air when the relative humidity rises above 55 per cent or the wet-bulb temperature rises above 60 F; also, if necessary, thermostat *D* shuts off the water supply to the spray heads in the air washer and opens the supply to the flooding nozzles at the eliminator plates, by operating the three-way valve *U* (Fig. 10). The relative humidity must, of course, be changed to suit the requirements. It must be maintained low enough to avoid condensation on walls or windows.[3]

5. Thermostat *E* in the discharge end of the air washer operates a three-way valve (*V*, Fig. 10) in the water circulating line so as to cause water to pass through or around a heating unit in order to produce the correct dew-point temperature by adding any necessary heat to the water. It may also operate a reverse valve *W* (Fig. 10) in the steam supply to the heating unit. The heat added may be only that sufficient to make up the temperature drop through the washer caused by evaporation. This thermostat is reverse-acting to prevent over-humidification in case of failure of the motive power.

6. Thermostat *F* in the fan discharge operates a valve in the steam supply to the heater *R* in order to produce the lowest temperature at which air can be introduced into the conditioned space, without complaints of draft. This varies from 60 F to 70 F, depending on the velocity through the supply grilles and their location.

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[3]See discussion of condensation in Chapter 7. Also see paper entitled Frost and Condensation on Windows, by L. W. Leonhard and J. A. Grant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

7. Room thermostat *G* in a representative location controls a valve in the steam supply to the coil or coils *S* which supply the heat to replace the loss from the conditioned space.

Summary Operation (With Refrigeration)

Thermostats *A*, *B*, *F*, and *G* all hold their valves closed during summer temperatures which are above the thermostat settings, although this is unimportant while no steam is being supplied.

1. Thermostat *C*, having been set for 50 F, supplies power to open wide the intake damper and close the return air damper under the higher summer temperatures, and this power can be passed through a graduating switch to permit manual operation of the dampers. As the wet-bulb temperature, or total heat, of the outdoor air is now normally greater than that of the return air, it is desirable in order to keep down cooling costs to recirculate the maximum amount of air.

2. Humidistat *D* is by-passed so that power is applied directly to the three-way valve *U* (Fig. 10) to prevent shutting off the sprays. This by-pass can be arranged for cutting in manually, or automatically, with the starting of the refrigerating machinery.

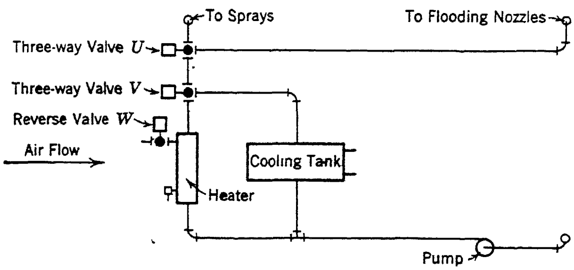

FIG. 10. CONTROL VALVES FOR A DEHUMIDIFYING AIR WASHER

3. Thermostat *E*, operating the three-way valve *V* (Fig. 10), now determines whether the spray water is to be passed through refrigerated coils or is to be recirculated without treatment, and thus it regulates the dew-point temperature. It is assumed that steam and refrigeration are not both turned on at the same time.

4. Thermostat *H* operates a damper *O* in the by-pass space around the air washer so as to mix the warmer return air with the cold air leaving the dehumidifier in such proportions as to give the minimum temperature at which air can be introduced into the conditioned space. This might be 70 F for a room temperature of 85 F. A switch installed in the power line from *H* should be so connected as to permit keeping damper *O* closed during winter operation.

5. Thermostat *G*, in addition to operating a valve on the heating unit *S*, acts as a pilot for thermostat *H* so as to retard the action of the latter in closing the by-pass damper when the temperature in the space is below the desired point.

Spring and Fall Operation

During a considerable part of the year, conditioning can be accomplished by using all outside air or by mixing it with returned air. For example, when the total sensible heat gain in an auditorium is 2.4 Btu per pound of air being treated, outside air will be raised from 60 F to 70 F by the heat gain. During this period when dry-bulb temperatures are to be maintained at, or not much above, 70 F, the gain in sensible heat is the only factor that need be considered, because it is large in comparison with the gain in latent heat, except in restaurants and in some classes of industrial work. The intake and recirculating dampers can then be operated by thermostat *F* set at 60 F. It is assumed that such an outlet

temperature can be used; if not, the volume of air should be increased. Thermostat *H*, being set higher for hot weather, holds by-pass damper *O* open to provide a maximum volume of air. In order to minimize over-humidification, the air washer and by-pass are arranged so that the return air stream tends to use the by-pass. However, since dehumidification is not required, the humidity control is obtained by shutting off the spray water by humidistat *D*.

Except for heating-up periods or other times when the heat gain is not greater than the heat loss, a system of this type can be operated without artificial heat with outdoor temperatures as low as 40 F. For this reason it is economical to place a thermostat in the return air near *D* set to shut off a valve in the main steam supply to the system at a temperature about 3 degrees below that desired in the conditioned space. A pilot thermostat exposed to the outdoor temperature prevents the shut-off on days colder than 40 F.

As previously stated, there can be many variations from these descriptions, some of which are:

1. Tempering coils may consist of only one bank, *P* or *Q*, controlled by thermostat *A* or thermostat *B*. In any case the capacity of the heating unit controlled by the outdoor temperature must be as low as feasible, otherwise if steam is supplied to it when the outdoor temperature is 30 F, the temperature of the air entering the washer is likely to be too high to permit maintaining the proper dew-point temperature.

2. Both tempering coils may be omitted and the return air may be mixed with outside air by thermostat *C* so as to provide a proper temperature at the washer inlet. In this case, humidistat *D* should not act as a pilot.

3. The heating unit for the air washer water may be omitted, and the proper dew-point temperature maintained by placing thermostat *C* in the location of *E*. This requires either additional heat from the tempering coils or more return air to make up the loss due to evaporation in the washer.

4. Heating unit *S* may be combined with *R* in one or two banks and controlled by a one- or two-point thermostat at *F*, set for the minimum temperature at which air can be admitted into the conditioned space. For heating purposes, thermostat *G* then becomes a pilot for *F* so that these heating units are operating at full capacity when the space is cold, and are throttled by *F* when no heat is required.

5. Another arrangement is the use of a type of thermostat at *F* which can operate at any temperature between a proper minimum and a necessary maximum, depending on the temperature of the space. Thus for winter operation when the room temperature is 68 F, the blower delivers air sufficiently warm to supply the heat required under extreme conditions, and when it is 74 F, the delivery will be as cool as possible without complaint of drafts. A similar device can be used to replace *H*, and be set to operate between 60 and 80 F for summer conditions.

6. For summer use, a remote readjustable thermostat can be located at *H*, and can be reset by a pilot exposed to the outdoor temperature. Thus as the outdoor temperature increases, the temperature in the space is maintained at a higher point.

7. A constant portion of the return air may be brought to a point between the air washer and the blower, and the temperature of the air leaving the washer may be regulated to give the proper result at *H*. The regulation is accomplished by shutting off one or more groups of sprays, or by changing the temperature of the spray water until the proper degree of cooling is secured.

8. Where an air washer large enough to pass all the air handled by the fan is selected, the by-pass and its damper *O* are not used. The washer sprays must be divided into two side-by-side sections so that one section can be turned on or off by *H* to provide the proper temperature.

9. Where an ejector type heating unit is used for the spray water, a reverse-acting valve similar to *W* (Fig. 10) must be placed in the steam supply to be operated by thermostat *E*. In this case it is usual to install in this steam line another reverse-acting

valve to be operated directly by the water pressure in the pump discharge line. This automatically shuts off the steam when the water circulating pump is not in operation.

10. Based on the fact that the spray water in the air washer pan has practically the same temperature as the air leaving the washer, dew-point control can be accomplished by installing thermostat *E* in the water pan.

11. Where cold well-water is used for dehumidification, it is admitted to the sprays through a three-way valve similar to *V* which is operated by thermostat *E*.

12. Control of steam heat is shown entirely by valves, although it is usual to install a by-pass damper around each heating unit and to operate it, either with or without a damper over the face of the heating unit, in conjunction with the valve.

PROBLEMS IN PRACTICE

1 ● How may temperature control be obtained in a room heated by a radiator with a constant steam supply?

By a thermostat handling an individual radiator valve pneumatically or electrically, or by a self-contained radiator valve.

2 ● How may temperature control be obtained in a room heated by a unit heater?

With constant steam supply, the unit heater motor may be started or stopped by a thermostat, either directly or through a relay. With intermittent steam supply, operation of the motor by thermostat can be limited to the time that steam is available, by using a reverse-acting temperature or pressure limit switch.

3 ● How may temperature control be provided in a room heated and ventilated by a unit ventilator which includes two extended-surface units?

Operation of the unit for service during occupancy of the room may be manual, by switch, or by time clock. When the desired temperature level is reached, the outside air intake may be controlled by a damper motor coupled with the fan motor circuit by means of a thermostat. The outside air damper will operate to a given position in either case.

Air passing through the unit may be preheated through the first heating coil to a definite temperature by a control valve on the steam supply governed by a temperature controller reacting to the temperature of the air on the outlet side of the convector. The second heating coil may provide the necessary heating capacity, and the steam supply to this coil may be modulated, either manually or automatically, in accordance with the temperature required in the room.

4 ● How may temperature control be obtained in a room heated by a duct system?

Air may enter the room from the central fan system at a predetermined minimum temperature. Heaters placed in the duct to bring the air up to this temperature should be equipped with face and by-pass dampers which may be adjusted by a positioning damper motor to give temperature control.

5 ● How may temperature be controlled in a room cooled by a unit cooler?

Practice indicates that a thermostat should provide for the automatic operation at all hours of the fan and control valve on the refrigeration source, but that there be a manual switch to enable the fan to operate continuously during occupancy.

6 ● How may temperature control be obtained in a room cooled by a self-contained mechanical unit?

The fan operation may be controlled by a manual switch, while a room thermostat in conjunction with a solenoid valve may regulate the flow of the refrigerant to the coil. The thermostatic circuit might be operative only when the fans are running; and the compressor might be controlled by refrigerant pressure.

7 ● How may temperature control be obtained in a room heated by an automatically-fired warm air furnace?

A room thermostat might control the combustion unit; and a limit switch in the top of the furnace unit, when at a low setting of its control might operate the fan whenever there is a rise of temperature, and when at a high setting of its control it might shut off the combustion unit. A room humidity control operating a solenoid valve on the water supply to the humidifier, or operating a relay on the recirculating pump motor to the humidifier, may be connected in parallel with the fan motor. Humidification may be supplied only when heat is supplied and when the humidity control acts in conjunction with a time switch.

8 ● How may humidity be controlled in a unit humidifier for a steam or hot water heating plant?

Since heat is required for evaporation, a temperature limit switch, preferably of the immersion type, may be placed in the heating supply riser to cause the unit to be inoperative when heat is not available. A room humidity control will operate a solenoid valve on the water supply to the sprays. Both the solenoid valve and the humidity control may be electrically wired in parallel with a fan motor, and be subject to the temperature limit switch.

9 ● Discuss a control system, including control of humidity, for the heating cycle of a central fan system of air conditioning.

During the heating cycle it is necessary to vary the amount of outdoor air drawn into the system in accordance with the temperature of that air. It is also advisable to adjust the volume of return air when mixing it with the outdoor air so that the resultant mixture will be of constant volume delivered to the preheater coils at some predetermined constant temperature.

By placing a temperature controller in the conditioner just ahead of the preheating coil, the temperature of the air delivered at that point may be measured, and by connecting this controller to a damper motor attached to the intake damper this damper can be operated by a temperature variation at the controller. The intake damper is so linked to the return damper that the combined volume of air delivered through the ducts of the system is constant. At a fall in outdoor temperature, this arrangement will move the intake damper to a closed position and the return damper to an open position, whereas the reverse will hold true when there is a rise in outdoor temperature.

If conditions prevent such mechanical linkage, it is possible to use two damper motors connected so they are operated individually but in inverse ratio.

The operation of the preheating coils should be dependent upon humidity conditions in the occupied spaces, and the humidity controller should be installed where conditions are representative of the humidity throughout the section, because air leaving the preheating coils is immediately passed through a spray where it becomes saturated with moisture. If the air is cold, it will absorb so little moisture that when it is delivered to the conditioned spaces its relative humidity will be low. When the compensated humidity control calls for additional moisture, the steam control valve in the preheater line should be opened to allow more steam to flow through the coils.

Whenever the preheating coils are being heated the spray should be in operation, but when the coils are cut off the air is sufficiently moist and the spray should be closed down. This necessitates an inter-connection between the control valve on the preheater and the spray pump on the water supply. Water is supplied to the spray during the heating cycle from a recirculating water tank beneath the sprays.

The reheating coil determines the dry-bulb temperature of the delivered air, so if the conditioner is equipped with both face and by-pass dampers on this coil it is obvious that these dampers should be controlled by a thermostat located at some representative position in the space being supplied with the conditioned air. If this thermostat is in turn connected with auxiliary apparatus which will vary the damper settings, it will be possible to pass more or less air through the reheater as the temperature falls or rises.

A low-limit temperature control might also be mounted in the discharge duct as a precaution against blowing cold air into the space. Such control would actuate the dampers of the reheater when the duct temperature fell below a predetermined minimum regardless of the demands of the master controller.

The amount of steam supplied to the reheater coils should be a function of the position of the dampers. If the face dampers are closed no heat is required, and to conserve steam suitable interconnection between the damper motor and the control valve should be made in order that this valve will close whenever the damper valve is closed. By adding modulating auxiliary apparatus to the steam valve, it may be made to operate proportionately to the setting of the dampers.

10 ● What is the relation between comfort, economy, and the use of temperature controls?

As a general rule, a moderate expenditure for control equipment can be justified on the basis of economy, but the cost of a complete system of individual room control can ordinarily be only partly so justified and the remainder must be charged to convenience and comfort. There are, however, many types of systems where the question would not arise, for without complete control equipment these systems would be unusable.

Chapter 15

AIR POLLUTION

Sources of Air Pollution, Effects of Air Pollution on Health, Pulmonary Effects, Occlusion of Solar Radiation, Industrial Air Pollution, Abatement of Atmospheric Pollution, Smoke Abatement, Dust and Cinder Abatement

THIS chapter considers the hygienic aspects of atmospheric pollution and the methods by which this pollution may be lessened. Information concerning the cleaning of air brought into buildings for ventilating purposes will be found in Chapter 16, and a discussion of the exhausting of dusts and toxic gases from factories and industrial plants is considered in Chapter 21.

The impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silk, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon. (See Fig. 1, Chapter 16, for size of impurities in air.)

Dust, Fumes, Smoke

The most conspicuous sources of atmospheric pollution may be arbitrarily classified according to the size of the particles as dusts, fumes, and smoke. *Dusts* are particles of solid matter varying from 1.0 to 150 microns in size. *Fumes* include particles resulting from chemical processing, combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size. *Smoke* is composed of fine soot or carbon particles, less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, sulphuric acid, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health.

The lines of demarcation in these three classifications are neither sharp nor positive, but the distinction is descriptive of the nature and origin of the particles, and their physical action. Dusts settle without appreciable agglomeration, fumes tend to aggregate, smoke to diffuse. Particles larger than one micron will eventually settle out by gravitation; particles smaller will remain in suspension as permanent impurities unless they agglomerate to sizes larger than one micron.

Fly-Ash, Cinders

The term *fly-ash* is usually applied to the extremely small particles of ash, and the term *cinder* to the larger particles of coke and ash which are discharged with the gases of combustion from burning coal.

AIR POLLUTION AND HEALTH

Many kinds of dusts and gases are capable of producing pathological changes which may cause ill health. The harmful effects depend largely upon the chemical and physical nature of the impurities, and the concentration, length of time, and conditions under which they are breathed. Dust particles must be minute in size to be inhaled at all, although fairly large particles may gain access to the upper air passages.

The human body possesses remarkable filtering media for protecting the lungs. Small hairs which line the nasal passages, and a multitude of microscopic hairs, called *cilia*, in the epithelial lining in the bronchial tubes intercept many of the dust particles before they reach the lungs.

The constant inhalation of dusts in city air irritates the mucous membranes of the nose, throat, and lungs, and eventually may produce discomfort and a series of minor respiratory disorders. The pigmented lung of the city dweller is an example of the pathological change produced over a period of years. This condition may be of no clinical importance, but an exaggeration of it in the coal miner results in anthracosis or dark spots on the lung due to the presence of pigment in the lymph channels which impairs the functioning of the lung cells under stress.

Effects of Solids

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it appears to have a marked influence in aggravating the disease once it has started. There is, however, quite reliable evidence that carbon pigment, one of the atmospheric dusts, tends to wall off local tuberculosis rather than to further its spread.

The sulphurous fumes and tarry matter in smoke are probably more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages, leading to asthmatic breathing and bronchitis and, in extreme cases, to death. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons. Physical examination showed difficult breathing, rapid pulse, cyanosis, cardiac dilation, and a redness and inflammation of the mucosa of the nose, mouth, throat, trachea, and bronchi.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough *CO*

to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is believed to be insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[1] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[2] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

The effect of air pollution on the health of city dwellers is difficult to determine, owing to the slowness of its manifestations. The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution have seemed justifiable to the public.

AIR POLLUTION IN INDUSTRY

In many industrial processes, sufficient amounts of dusts, fumes, and vapors are liberated to be injurious to the health of workers. Some dusts are poisonous (lead, mercury, arsenic, manganese, and cadmium) and some act as irritants (silica, steel, iron, and granite). Certain dusts may produce catarrhal conditions and increase susceptibility to such diseases as bronchitis, pneumonia, and tuberculosis. Silicious dust is especially harmful because it has a direct damaging action upon the tissue of the lungs, but organic dusts, both animal and vegetable (hair, pollen, textile, and fiber), do not seem to affect the lungs at all, although they may cause considerable discomfort in the upper respiratory passages to persons sensitive to them.

Industrial gases and fumes act specifically upon the mucous membranes, the lungs, blood, skin, and eyes. Some extremely poisonous gases act after very short exposures. Among these are carbon monoxide, hydrogen sulphide, ammonia, chlorine, bromine, arsine, and cyanogen.

The industrial processes which liberate harmful substances are too manifold and the effects too diverse to be considered here, where discussion is limited to the commonest and most serious with which the ventilating engineer may be confronted, namely, carbon monoxide, lead, and silica. For a more thorough treatise on the subject reference should be made to books by Hamilton[3], Rosenau[4], and Henderson and Haggard[5].

Carbon Monoxide Poisoning

Carbon monoxide is a common form of poisonous industrial gas, met with in mines, foundries, coke-oven sheds, garages, and houses. Its action

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[1] Effects of Atmospheric Pollution upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).
[2] Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin No 197*, 1930).
[3] Industrial Poisons in the United States, by Alice Hamilton.
[4] Preventive Medicine and Hygiene, by Milton J. Rosenau.
[5] Noxious Gases, by Y. Henderson and H. Haggard.

is due to the fact that the combining power of carbon monoxide with the haemoglobin of the red blood corpuscles is about 300 times greater than that of oxygen. Since the resulting stable combination destroys the power of the haemoglobin to unite with oxygen in the lungs and to supply it to the tissues, the effects are due to lack of oxygen, and the symptoms are those of anoxemia, namely, dizziness, headaches, sleepiness, fatigue, and, in extreme cases, paralysis and death. The dangerous saturation level of the blood with carbon monoxide is about 50 per cent. Even as little as 0.07 per cent in the air will render, in half an hour, one quarter of the red corpuscles incapable of uniting with oxygen. One to two parts per 10,000 parts of air is set as a safe limit of pollution which may be breathed for a long time without producing perceptible symptoms.

Silicosis

Silicosis is a chronic disease of the lungs which results from the local physio-chemical action of hydrated silica upon the pulmonary tissue, causing progressive lymphatic fibrosis, and rendering the tissue susceptible to tuberculosis. The disease is slow in evolution, requiring usually a number of years of exposure. It occurs principally among granite workers, sand blasters, metal miners, metal polishers, potters, and millstone workers.

Lead Poisoning

Lead poisoning is the most insidious and most common of all industrial diseases. It occurs principally among lead workers and smelters, lead miners, potters, painters, typesetters, stereotypers, plumbers, and workers with glass, gold and silver. Lead, in practically all forms, is a cumulative poison which is absorbed by way of the blood stream, chiefly from the respiratory tract, but also from the digestive tract and from the skin. The effect may be either an acute or chronic poisoning. The principal symptoms are colic, constipation, anemia, headache, anorexia, a bluish line along the edges of the gums, rheumatic pains, and, in extreme conditions, paralysis, blindness, insanity, and death.

It has been found[6] that 2 mg per day is the smallest dose, by inhalation, which in the course of years may result in lead poisoning. Regular inhalation during the usual working hours of air containing less than 0.2 mg of lead per cubic meter does not seem to produce serious lead poisoning in individuals of representative industrial groups[7].

Prevention

The prevention of industrial hazards from dusts and poisonous gases is largely a ventilation problem consisting of keeping the impurities in air down to a safe concentration. As yet there are no generally accepted standards on which to base the design of the ventilation equipment. Approximate data on the toxicity of various gases and fumes met with in industrial establishments are given in Table 1. Column 5, giving the maximum allowable concentrations for prolonged exposures, was compiled from experiments in which most exposures lasted not more than a

[6]Lead Poisoning, by Thomas Morrison Legge (*Journal Royal Society Arts*, 1929, Vol. 77, p. 1023).

[7]What is a Dangerous Quantity of Lead Dust in Air, by C. M. Salls (*Industrial Hygiene Bulletin*, New York State Department of Labor, 1925).

week, and it is reasonable to assume that over more prolonged exposures such concentrations would cause pernicious effects.

Much is known concerning the physiological and pathological effects induced by various types and concentrations of atmospheric pollutants. In the absence of an accepted standard for safe breathing, and because of the slow, cumulative effects of certain kinds of air contaminants, the best procedure is the periodic medical examination of individuals, and the

TABLE 1. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR[a]

VAPOR OR GAS	RAPIDLY FATAL	MAXIMUM CONCENTRATION FOR FROM ½ TO 1 HOUR	MAXIMUM CONCENTRATION FOR 1 HOUR	MAXIMUM ALLOWABLE FOR PROLONGED EXPOSURE
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	....	....	....
Hydrocyanic acid	30	1½	½	⅕
Ammonia	50–100	25	3	1
Hydrochloric acid gas	10–20	½	....	1/10
Chlorine	10	½	....	1/100
Hydrofluoric acid gas	2	1/10	....	1/33
Sulphur dioxide	4–5	½–1	....	1/10
Hydrogen sulphide	10–30	5–7	2–3	1
Carbon bisulphide	....	11	5	½
Phosphene	20	4–6	1–2	....
Arsine	2½	½	½	....
Phosgene	Over ¼	¼	....	1/100
Nitrous fumes	2½–7½	1–1½	....	⅓
Benzene	190	....	31–47	1½–3
Toluene and xylene	190	....	31–47	....
Aniline	....	....	1–1½	1/10
Nitrobenzene	....	....	1/100	1/500
Petrol	243	100–220	....	....
Carbon tetrachloride	480	240	40	16
Chloroform	250	140	50	2
Tetrachlorethane	73	....	....	1½
Trichlorethylene	370	....	....	....
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead vapor	....	....	....	5–6

[a]Original data compiled by Y. Henderson and H. Haggard. (See *Noxious Gases*, 1927.) Data revised by T. M. Legge. (See *Lessons Learned from Industrial Gases and Fumes*, Institute of Chemistry of Great Britain and Ireland, London, 1930.)

routine measurement and study of the concentration and the physical and chemical characteristics of the dusts to which those individuals are exposed.

ABATEMENT OF SMOKE AND AIR POLLUTION

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is

the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 27.)

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner fuel beds* on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 40), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the

smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide and sulphuric acid mist, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in later paragraphs.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coal when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

Dust and Cinder Catchers[8]

The various types of dust and cinder catchers available today can be divided into six general classes:

1. Settling chambers.
2. Dust and cinder traps.
3. Centrifugal separators.

[8]See Smoke and Dust Abatement, by M. D. Engle (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

4. Electrostatic precipitators.
5. Gas scrubbers.
6. Fabric filters.

The selection of the proper type of catcher calls for a careful study of the material to be caught and the draft and space available. After installation, constant vigilance is necessary to keep the catchers in proper working condition if satisfactory operation is to be obtained.

If possible, the dust or cinder catcher should be installed on the inlet side of the induced draft fans because the dust and cinders in the gases seriously erode the wheels of the fans, the inlet connections and the scrolls. Where the induced draft fans operate at high tip speeds and no catchers are installed, it is not uncommon for the fans to require major repairs within one year and complete replacement within five years.

Settling Chambers

Probably the oldest form of dust catcher is the settling chamber, which generally consists of a large-sized, gas-tight space into which the dust-laden gases are discharged before being delivered to the chimney. The velocity of the gas should be reduced to a point where the larger and heavier particles will be precipitated by gravity. For good operation, the velocity of the gas should be reduced to a maximum of 2 fps. The bottoms of the chambers should be provided with dump plates through which the collected dust can be removed. Because these chambers are not effective in removing the finer dust particles they have been practically superseded by smaller and less costly devices.

Traps, Catchers, Precipitators

Various types of traps have been devised. In general they all depend upon breaking the gas up into thin strata and subjecting those thin strata to several abrupt changes in direction. The dust is thrown out of the gas stream into specially shaped pockets, or impinged against a roughened surface. The trapping pockets are drained into a hopper below with a small quantity of gas and the dust settles out by gravity due to the low velocity in the hopper. In the roughened surface type, various sections of the trap are closed off at intervals by means of dampers and the dust is shaken off the roughened surface into a hopper below.

These devices work very well in catching large size dust and cinders and trap much of the fine dust. They have been used most extensively on stoker-fired installations. They have the advantages of low pressure drop, relatively small space requirements, and low first cost.

Centrifugal catchers obtain separation by projecting the particles tangentially out of the gas stream. The effectiveness of this type of catcher varies directly as the specific weight of the dust and as the square of the tangential velocity, and inversely as the radius of rotation.

Electrostatic precipitators are used for catching fine dust. These precipitators consist of dust-tight chambers in which are suspended reinforced concrete slabs on about 10-in. centers. Between the slabs are suspended bare metal rods. High-voltage unidirectional current is applied to the reinforcing rods in the concrete slabs acting as positive electrodes, the bare rods acting as negative electrodes. The dust-laden

gas flows horizontally through the precipitator and the dust particles migrate toward the concrete slabs to which they adhere and then fall or are scraped off into the dust hoppers below.

Gas Scrubbers

Wet scrubbers have been used for many years for removing dust from gases. A number of different types of scrubbers are now being built for removing dust from boiler flue gases. One type depends upon saturating the gas and washing the dust out of suspension by a spray of water. For best results with this type, the water should be atomized into as fine a spray as possible.

Another type depends upon splitting the gas into thin strata and subjecting these strata to a number of abrupt changes in direction, throwing the dust against the wet surfaces. The main problem in developing a satisfactory wet dust catcher is to find suitable materials of construction that will resist the corrosive action of the wash water for a reasonable length of time.

Fabric Filters

Filters of many kinds have been used with variable success. The filter bags are made of cotton, wool or asbestos fabric. The fabrics used in these filters do not withstand the temperatures at which gases are usually discharged from the boilers, and hence the gases must be cooled by some means. Surface coolers or water sprays can be used for reducing the gas temperatures.

One of the serious objections to all of these dust catchers is the relatively high cost of installation and maintenance, and the space required for installation.

Disposal of Dust and Cinders

Even after the dust and cinders have been caught, the disposal of the material caught presents a serious problem. The cinders discharged with the gases from stoker-fired boilers are usually very high in carbon and contain from 50 to 80 per cent as much heat per pound as the coal which is being burned. It is possible, and usually economical, to burn these cinders. They cannot be satisfactorily mixed with the coal in the stoker hopper but they can be blown into the furnace over the stoker fuel bed and burned satisfactorily. If a sufficient quantity of cinders is caught, a small unit pulverizer can be installed to prepare them for burning over the stoker fuel bed. The same pulverizer can be used for coal at times of peak load and will materially increase the capacity of the fuel-burning equipment for the boiler to which it is connected.

No satisfactory market has been developed for the dust caught from pulverized coal installations, but the possibilities are being investigated and it seems likely that in the future this material will have a market value that will go a long way toward paying the fixed charges on the cost of catching it.

The distribution of dust in the gas entering and leaving the dust and cinder catchers is not uniform and is different in practically every in-

stallation, and varies widely with changes in furnace conditions. In order to obtain a representative sample it is necessary to traverse the inlet and outlet of the catcher with a sampling tube which faces into the gas flow. The velocity of the gas into the sampling tube must be the same as the velocity of the gas in the duct at the instant the sample is taken. The swirls and eddy currents in the ducts make it difficult to obtain consistent readings, but if the test is conducted by some one of experience, an indication of the approximate efficiency can be obtained.

Nature's Dust Catcher

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. In fact, without dust in the air to form the nuclei for rain drops it would never rain, and the earth would be continually enveloped in a cloud of vapor.

PROBLEMS IN PRACTICE

1 ● What is a micron?

A micron equals 0.001 millimeter or approximately $\frac{1}{25000}$ in.

2 ● Distinguish between dusts, fumes, and smokes.

Solid particles ranging in size from 1.0 micron to 150 microns are called *dusts.*

Particles resulting from sundry chemical reactions and ranging from 0.1 to 1.0 micron in size are called *fumes.*

Carbon particles less than 0.1 micron in size which generally arise from the incomplete combustion of such materials as coal, oil, or tobacco are called *smokes.*

3 ● What are some of the more important physical properties of these various groups of foreign bodies which are of importance in ventilation?

In slowly moving air, dusts tend to settle out by gravity without agglomerating to form larger particles; fumes have the tendency to form larger particles which will settle when they attain the size of approximately 1.0 micron; while smokes tend to diffuse and remain in the air as permanent impurities.

4 ● Why is atmospheric pollution an important engineering problem?

a. Certain impurities, when present in too great concentrations, cause ill health or even death.

b. High concentrations of solids occlude solar radiations.

c. Some materials cause permanent injury to parts of buildings, as sulphur fumes corrode exposed metal.

d. Extra cleaning expense is incurred in dusty localities.

e. Internal combustion engines are damaged by abrasive dusts.

5 ● How may the hazards of dust-producing industrial operations best be curtailed?

By providing mechanical exhaust ventilation sufficient to keep dust concentration at a safe level (see Table 1) and then removing foreign bodies to reduce the pollution of outside air.

6 ● How may the pollution of the atmosphere be lessened?

By compelling industrial plants to install dust catching and smoke controlling devices. In many cities the domestic heating plant is one of the most serious offenders, but these

plants are too small to justify the installation of dust catchers. Public education in improved firing methods would be of considerable help in this field.

7 ● Compare the dry and wet types of dust catchers.

The dry types are very effective in removing the larger dust particles but the smaller particles generally pass through other kinds than the electric precipitator. The dry types also require considerable space and therefore sometimes introduce resistance to the flow of air. The wet types are effective in removing some of the smaller dusts and the water-soluble gases. The principal disadvantage of the washer is its short life caused by the corrosive action of the wash water.

8 ● What size particles are detrimental to health?

While fairly large particles may enter the upper air passages, those found in the lungs are seldom more than 10 microns in size, and comparatively few of them are more than 5 microns. It is agreed that particles between ½ and 2 microns may be harmful; some authorities place the upper limit at about 5 microns, and some incline to extend the lower limit to 0.1 of a micron.

9 ● Is the shape of the particle of any significance?

Hard particles with sharp corners or edges have a cutting effect on the delicate mucous membranes of the upper respiratory tract which may lower the resistance of the nose and throat to acute infections. This is aggravated by the irritating effects of some chemical compounds which may be taken in with the air and which act to reduce resistance.

10 ● What are the principal meteorological effects of smoke and dust?

a. The reduction in the amount of light received. Measurements have shown that visible light may be as much as 50 per cent less intense in a smoky section of a city than in a section that is free from smoke. Ultra-violet light is reduced as much or more, and in some cases is cut out entirely for a time.

b. Smoke and dust aid in the formation and prolongation of fogs. City fogs accumulate smoke and become darker in color and very objectionable. The sun requires a longer time to disperse them, and when the water is evaporated, there is a rain of smoke and soot particles that have been entrained.

11 ● Why has not smoke abatement been more effective?

Because communities have not been made sufficiently aware of the possibilities of burning high volatile fuels smokelessly and of separating cinder and ash from the stack gases to a degree that will prevent a nuisance.

12 ● Is the abatement of dust and cinders important?

Yes. Only a small percentage of the solid emission from stacks is smoke, in the accepted popular sense; the remainder is fly-ash and cinders. While black smoke is disagreeable and its tarry matter and carbon particles soil anything with which they come in contact, the cinders and some of the ash are hard and destructive. They also, together with dusts from industrial processes, make up the hard, sharp, irritating, air-borne solids that are breathed by individuals not working in a dusty mill or factory.

13 ● Are air-borne impurities causative factors in hay fever, bronchial asthma, and allergic disorders?

Yes. Recent medical investigations indicate that 90 per cent of seasonal hay fever and 40 per cent of bronchial asthma are caused by air-borne pollens, tree dusts, and other allergic irritants.

14 ● Name some essential requirements for the smokeless combustion of fuels.

Time, temperature, and turbulence. A study of these factors is usually of value in overcoming a smoke nuisance.

15 ● What is the Ringelmann Chart Method of comparing smoke densities?

See Chapter 40. The Ringelmann Chart consists of four cards ruled with lines having different degrees of blackness. These cards, together with a white card and a black one, are hung in a horizontal row 50 ft from the observer. At this distance the lines become invisible and the cards appear to be different shades of gray, ranging from white to black. The observer, by matching the cards against the shades of smoke coming from a stack, is able to estimate the blackness of the smoke as compared with the chart.

Chapter 16

AIR CLEANING DEVICES

Requirements of an Air Cleaner, Types, Air Washers and Scrubbers, Viscous Type Filters, Dry Air Filters, Air Filter Installations

THE removal of impurities from air brought into a building for ventilating or air conditioning purposes is the function of any air cleaning or filtering device. These impurities include carbon (soot) from the incomplete combustion of fuels burned in furnaces and automobile engines, particles of earth, sand, ash, automobile tires, leather, animal excretion, stone, wood, rust and paper, threads of cotton, wool and silk, bits of animal and vegetable matter, bacteria and pollen. Microscopic examination shows that the character of the impurities varies with the locality, but as a rule carbon forms the greater part of them while the total is somewhat proportional to the state of industrial activity and the wind intensity. Additional information on sources of air pollution will be found in Chapter 15.

Observations have shown that practically all atmospheric impurities are less than 5 microns in size. (One micron equals 0.001 millimeter or approximately 0.00004 in.) The size and composition of each individual particle determines its buoyancy and consequently the length of time it will remain in suspension. The chart, Fig. 1, shows graphically the sizes of impurities found in the air, and other related data.

To estimate the probable dust load for air filter installations, the following approximate averages of atmospheric dust concentration may be used (7000 grains equal 1 lb):

Rural and suburban districts	0.2 to 0.4 grains per 1000 cu ft
Metropolitan districts	0.4 to 0.8 grains per 1000 cu ft
Industrial districts	0.8 to 1.5 grains per 1000 cu ft

REQUIREMENTS OF AN AIR CLEANER

To fulfill the essential requirements of clean air, an air cleaner should:

1. Be efficient in the removal of harmful and objectionable impurities in the air, such as dust, dirt, pollens, bacteria.
2. Be efficient over a considerable range of air velocities.
3. Have a low frictional resistance to air flow; that is, the pressure drop across the filter, measured in inches of water, should be as low as possible.
4. Have a large dust-holding capacity without excessive increase of resistance, or have ability to operate so as to keep the resistance constant automatically.
5. Be easy to clean and handle, or clean itself automatically.
6. Leave the air passing through the cleaner free from entrained moisture or charging liquids used in the cleaner.

The A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilating Work[1] explains how such devices are rated by (1) capacity in cubic feet of air handled per minute, (2) resistance

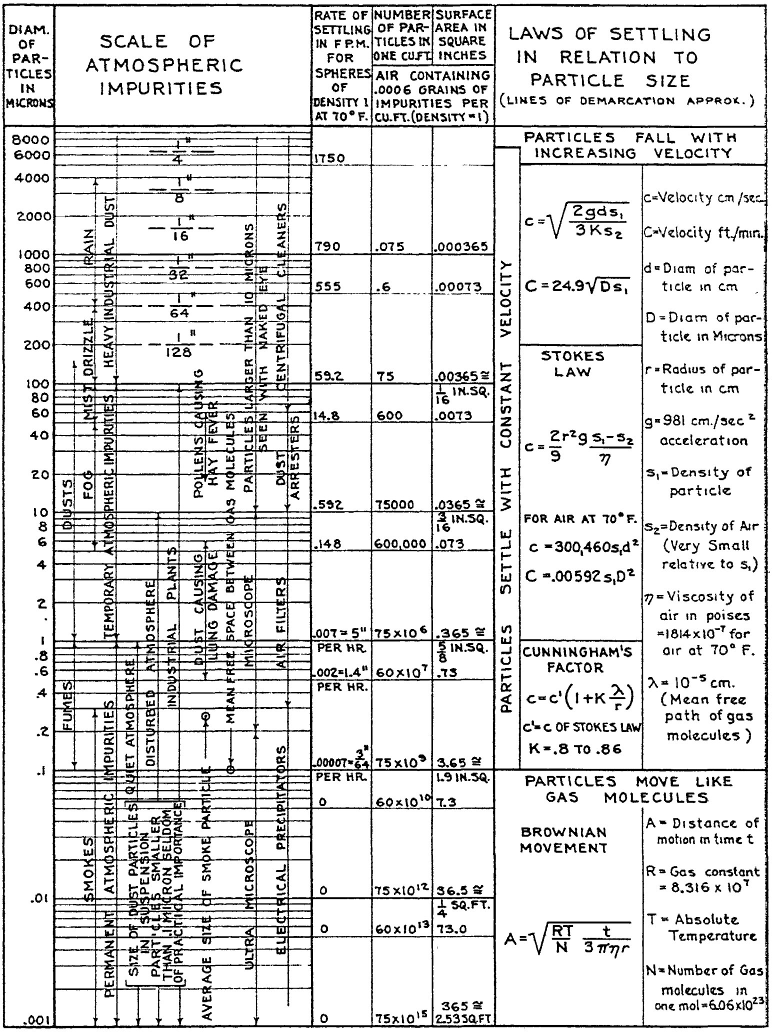

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FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

in inches of water at rated capacity, (3) dust arrestance, the percentage relationship expressing dust removal efficiency at rated capacity, (4) reconditioning power, the energy necessary to operate the mechanism of

[1]Adopted 1934 by A.S.H.V.E. See Chapter 41.

an automatic air cleaning device, and (5) dust holding capacity, the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

TYPES OF AIR CLEANERS

According to the Code, the following four classifications are given the devices:

Class A. Automatic Type: In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

Class B. Low Resistance Non-Automatic Type: Air cleaning devices for warm air furnaces, unit ventilating machines and similar apparatus and installations in which a maximum of not more than 0.18 in. water gage is available to move air through the air cleaning device.

Class C. Medium Resistance Non-Automatic Type: Air cleaning devices for systems in which a maximum of not more than 0.5 in. water gage is available to move air through the air cleaning device.

Class D. High Resistance Non-Automatic Type: Air cleaning devices for the air intake of compressors, internal combustion engines, and the like, where a pressure of 1.0 in. or more water gage is available to move air through the air cleaning device.

Air cleaners may be also classified as follows:

1. According to principle of air cleaning.
   a. Air washers.
   b. Viscous air filters.
      (1) Unit type.
      (2) Automatic type.
   c. Dry air filters.
2. According to application.
   a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type are usually recommended and are installed in a central plenum chamber.
   b. For unit ventilators. Filters of viscous unit or dry type, installed at inlet of individual units.
   c. For window installations. Self-contained units consisting of fan and filter, usually dry type, adapted to be placed in the ordinary window.
   d. For warm-air furnaces. Unit type viscous or dry filters placed in small plenum chamber of warm-air house heating systems.
   e. For compressors and Diesel engines. Unit type viscous or dry filters, installed at air intake of compressors and Diesel engines.
   f. For compressed air lines. Unit type viscous or dry filters.

With the growing congestion of large cities and an industrial growth throughout the entire country, the percentages of foreign material in the air, such as soot or carbon, which are unaffected by an air washer type of air cleaner, have increased. This has brought about the development of the viscous and dry type air filters which are part of many ventilating and air conditioning systems.

AIR WASHERS AND SCRUBBERS

Information on air washers will be found in Chapter 11.

Scrubbers have not been used very extensively in the past for cleaning

air for ventilating purposes. However, new types have been developed which appear to have possibilities for cases where the air to be cleaned is extremely dirty or where a higher degree of cleanliness is desired than can be obtained with an air washer.

VISCOUS TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement*. Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. While the arrangements of filtering media and the kind of materials used are almost unlimited, there are certain rather definite requirements for a practical commercial filter.

Investigations in this country and abroad demonstrate that the first impingement of dust laden air on a viscous coated surface removes about 60 per cent of the dust, the next impingement takes 60 per cent of what then remains—that is, 24 per cent—and the next impingement removes 9.6 per cent. To secure maximum efficiency, it is necessary to divide the air into innumerable fine streams, as the more intimately and freely the air is brought into contact with the viscous-coated media the better will be the cleaning.

The binding liquid used with viscous filters should have the following properties:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.
2. The viscosity should vary only slightly with normal changes of temperature.
3. It should be germicidal in its action to prevent the development of mold spores and bacteria on the filter media.
4. The liquid should flow freely at low temperatures.
5. Evaporation should not exceed 1 per cent.
6. It should be fireproof.
7. It should be odorless.

Viscous Unit Filters

In the unit type viscous filter, the filtering media are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet. The necessary washing, draining, and recharging equipment should be installed near each group of unit filters, with hot water and sewer connections provided.

To secure greater dust holding capacity and a practically constant resistance and air volume, the filter media are usually placed in the direction of air flow, with progressively finer filter densities determined by the percentage of dust impinged. This arrangement provides relatively large spaces for the collection of dirt in the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while at the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impinge-

ment type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency of the unit is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity of a built-up filter may be held at any desired figure. The frequency of cleaning any unit filter installation depends upon the dust concentration

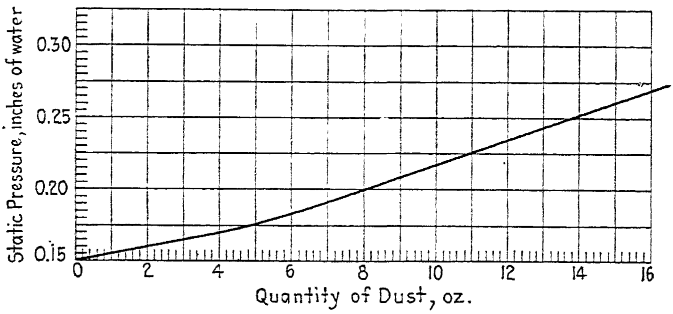

FIG. 2. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

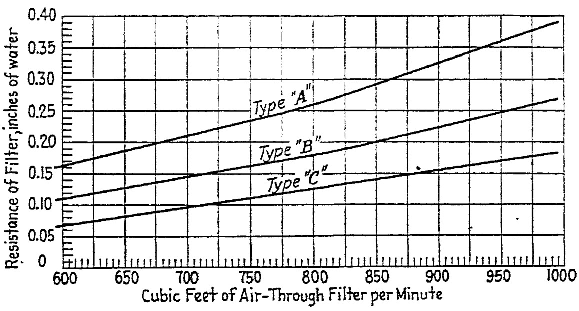

FIG. 3. RESISTANCE TO AIR-FLOW OF A TYPICAL UNIT AIR FILTER

of air being cleaned, and on the amount of dirt which can be accumulated in the filter medium without causing excessive resistance.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool or steel wool are available. Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm air furnaces and other installations where first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

The principle of air cleaning used in the viscous automatic filters is the same as in the unit filters. The removal of the accumulated dust,

however, is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter thus is deposited finally in the bottom of the viscous fluid reservoir from which it may be removed by different methods, depending on the design of the filter.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used by each type, as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain rotates slowly through this bath, thus performing the cleaning and recoating of the filter medium.

2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.

3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

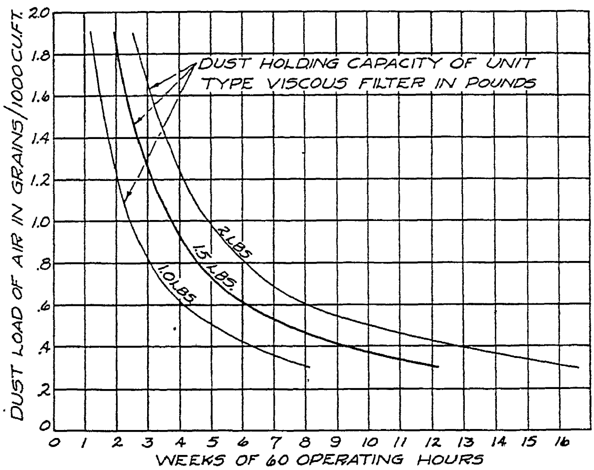

FIG. 4. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

The washing and renewing process in automatic filters usually is intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is ⅜-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or filtered through screens made of felt, cloth, or cellulose, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture

of the filtering media used in most of the dry filters, the surface velocity, or velocity of the air entering the media, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, and the filter media are usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As in viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering media are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials as filter media depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter media should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

INSTALLATION METHODS

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 per cent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.
2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.
3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.
2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.
3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.
4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.

6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.

7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.

8. Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

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PROBLEMS IN PRACTICE

1 ● What is meant by air filter performance characteristics?

The factors that determine the performance of an air filter, which are:

(1) efficiency in dust removal, (2) operating resistance, (3) dust holding capacity. In a properly designed filter these factors are balanced to obtain the desired characteristics for a given application. Since the requirements vary for different kinds of air cleaning service, it is necessary to have filters of different types to meet the various conditions.

2 ● What are the advantages of viscous filters?

The principal advantage of the viscous filter is its large dust holding capacity. The dust accumulation is distributed through the depth of the filtering medium rather than upon the surface as in the dry types, which makes it possible for viscous filters to handle heavy dust concentrations without excessive resistance. Since its efficiency and resistance are based on maximum air velocities of from 300 to 500 ft per minute through the filter, the viscous filter consumes the minimum amount of space for a given air volume.

3 ● What are the advantages of dry filters?

Dry filters are more efficient in the removal of fine dust particles from the air, and some types will eliminate even as much as 60 per cent of the smoke particles. Dry filters also are easily and conveniently maintained by vacuum cleaning, vibrating, or renewing the filtering medium.

4 ● If an air washer is used for cooling and humidity control in an air conditioning system, is a filter needed?

An air filter is desirable in conjunction with an air washer because of the large amount of soot in the air which, due to its greasy and amorphous nature, is not readily trapped in

an air washer. Filters should be placed between the washer and the air intake so that all the dirt will be collected at one point to simplify maintenance, to protect all the equipment in the system, and to prevent contamination of the water used in the washer.

5 ● Is an air filter needed with an extended surface type heat exchanger?

An air filter is essential with an extended surface heat exchanger in order to maintain its efficiency, for without this protection dust particles will adhere to the exposed surfaces, and gradually build up a deposit to the point where the efficiency will be impaired and the resistance increased by restricting the air passage.

6 ● What is the proper location of a filter in relation to the fan?

A filter will operate equally well whether placed on the suction or discharge side of the fan. It has become standard practice, however, to locate the filter on the fan inlet side because there it has: (1) simpler duct connections, (2) reduced static pressure losses, (3) more even air distribution over the entire filter area. Where an exceptionally high efficiency in dust removal must be maintained, it is often advisable to place the filter on the discharge side of the fan so there can be no infiltration of unclean air.

7 ● What instruments and apparatus are required for determining the pollen concentration in air by means of the settling method?

A microscope with a field of know area and a glass slide coated with a viscous material.

8 ● Describe the procedure for determining the pollen concentration in air by means of the settling method.

A glass slide coated with a viscous material is placed for a period of 24 hours in a horizontal position in the atmosphere to be tested. The slide is then removed and placed under the microscope, and pollen counts are made of approximately 25 fields over the area of the glass slide. Having determined the count over a definite area, as for example, 1 sq cm, and finding the settling rate of the average particles from the chart, Fig. 1, the concentration in parts per cubic yard can be calculated.

9 ● The resistance to air flow of a unit air filter is found to be 0.4 in. of water. The volume of air passing through the filter is 1000 cfm at a velocity of 200 fpm. What would be the filter area required in order to reduce the pressure drop across the filter from 0.4 in. of water to 0.16 in. of water?

Referring to Fig. 3: The resistance is substantially proportional to the square of the velocity, or

$$\frac{R_1}{R_2} = \frac{V_1^2}{V_2^2}$$

$$\frac{0.4}{0.16} = \frac{200^2}{V_2^2}$$

$$V_2^2 = 16{,}000$$

$$V_2 = 126.5 \text{ fpm}$$

$$Q = AV$$

$$1000 = 126.5\,A$$

$$A = \frac{1000}{126.5} = 7.91 \text{ sq ft}$$

The filter area would be increased from 5 sq ft to 7.91 sq ft.

10 ● A ventilating system complete with filters has a fan which, when operating at 400 rpm and delivering air at 1 in. of water total static pressure, requires an input of 3 horsepower. After the system operates for a time, the pressure drop across the filter caused by the clogging action of the collected dust and dirt increases from 0.1 in. of water to 0.4 in. of water. To maintain the original

rate of air delivery with the increased static pressure, at what speed must the fan be run and what horsepower will be required?

Static pressure after clogging of filter = 1 + (0.4 − 0.1) = 1.3 in. of water.

The static pressure varies as the square of the fan speed. Therefore, if X is the fan speed after the static pressure increases:

$$\frac{1.3}{1} = \left(\frac{X}{400}\right)^2$$

$$X = 456 \text{ rpm.}$$

The horsepower varies as the cube of the fan speed. Therefore, if Y is the horsepower after the static pressure increases:

$$\frac{Y}{3} = \left(\frac{456}{400}\right)^3$$

$$Y = 4.44 \text{ horsepower.}$$

To maintain the original rate of air delivery with the increased static pressure, the fan speed must be increased from 400 to 456 rpm, and the horsepower from 3 to 4.44.

Chapter 17

FANS AND MOTIVE POWER

Performance, Fan Efficiency, Characteristic Curves, Selection of Fans, Controls, Designation of Fans, Motive Power, Electric Power

FANS are used for producing air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where an axial flow fan is intended for operation at comparatively high pressures the hub sometimes is enlarged in the form of a disc and the fan is known as a *disc fan*.

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan: a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances, especially where no long ducts or heavy friction must be overcome and where noise is not objectionable, whereas centrifugal fans are commonly employed for operation at the comparatively higher pressures and where extreme quietness is necessary.

PERFORMANCE OF FANS

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These

laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and power?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.07495 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.06018 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.06018}{0.07495} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.06018}{0.07495} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and power vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and power?

$$\text{Speed} = 400 \times \sqrt{\frac{0.07495}{0.06018}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.07495}{0.06018}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.07495}{0.06018}} = 4.46 \text{ hp}$$

6. For a constant weight of air:
   (*a*) The speed, capacity, and pressure vary inversely as the density.
   (*b*) The horsepower varies inversely as the square of the density.

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and power?

$$\text{Speed} = 400 \times \frac{0.07495}{0.06018} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.07495}{0.06018} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.07495}{0.06018} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.07495}{0.06018}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the power required in moving the air to the power input to the fan. The work done in moving the air may be computed on the basis of either the static or the total pressure. When the static pressure is used in the computation it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static efficiency}^1 = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6369 \times \text{power input expressed in units of 746 watts}} \quad (1)$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total or dynamic pressure. This efficiency is variously known as the total, dynamic, or mechanical efficiency, and may be expressed as follows:

$$\text{Total efficiency} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6369 \times \text{power input expressed in units of 746 watts}} \quad (2)$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different percentages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape

[1]See Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, Edition of 1932.

and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[2] as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers*. The results of tests are plotted in different ways: the abscissae may be the ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this

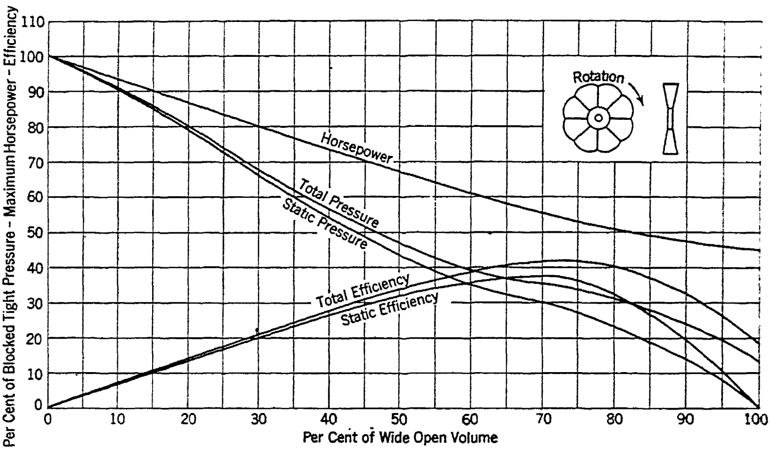


FIG. 1. OPERATING CHARACTERISTICS OF AN AXIAL FLOW FAN

chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. The proportions of the housing also affect the performance. As a rule a narrow fan of large diameter shows a higher efficiency than one of greater width and smaller diameter. For a number of designs using blades of certain shapes the proportion of the width to the diameter is so definitely established by the service for which the fan is intended that little variation in efficiency occurs, but in other designs, particularly that which uses straight radial blades, the efficiency may vary over a wide range depending on whether the dimensions are suitable for a fan intended for ordinary ventilating purposes or for a pressure blower. Figs. 1 to 4 show characteristic curves for different types of fans

[2]A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923. Amended June, 1931.

using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and more particularly to show the manner in which variations occur with changes in fan capacity.

Axial flow fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable

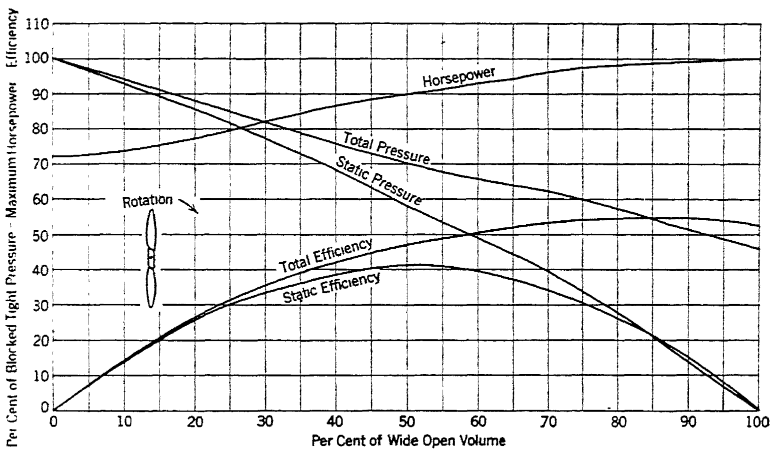


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in power as the resistance increases. The low efficiency when overcoming heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the pressure curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch,

camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

Another advantage of this type of axial flow fan is its low resistance to air passage when standing still. There are some installations in which such a characteristic is desirable.

The straight blade (paddle-wheel) or partially backward curved blade type of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static

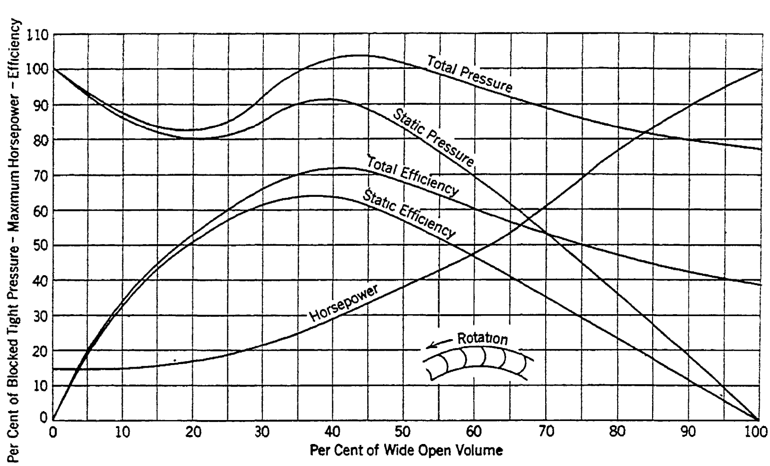


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity, and is quiet in operation. The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. Fig. 3 shows that this type of fan will have both a high and a low delivery for a given static pressure at constant speed. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4.) This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for

like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type makes it adaptable for direct connected electric motor drives. The high speed may necessitate somewhat heavier construction and more operating attention or service. The dimensional bulk for a given duty often is 150 per cent of that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type centrifugal fans a number of modified designs exist, differing in the

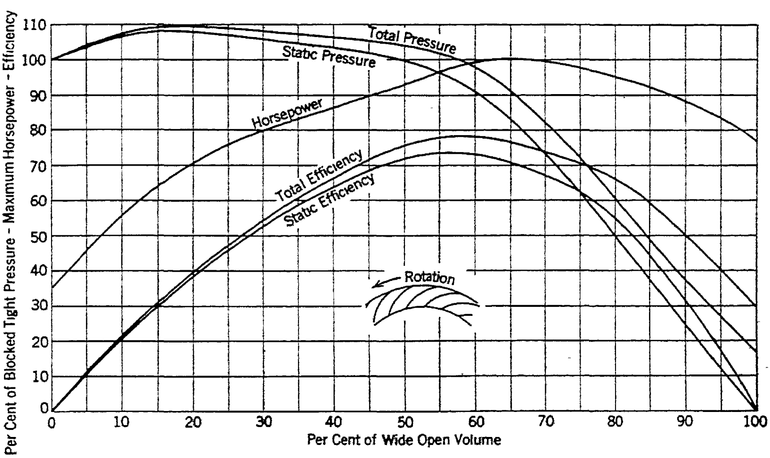


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade type with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show varying degrees of resemblance to the curves of Figs. 3 and 4, according to the degree of similarity to one or the other of the two designs of fan considered.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) speed, (3) noise, (4) size and weight, and (5) cost.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures:

1. Volume of air in cubic feet per minute (68 F, 50 per cent relative humidity, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake power.
5. Tip or peripheral speed.
6. Static pressure.

The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Fans for Ventilation and for Cooling Systems

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Because noise in a ventilating system is irritating and a cause for complaint, fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls, and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed required for a specified capacity and pressure varies with the type of blade, and that a tip speed which may be excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan is selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed in Tables 1 and 2 for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with the design of wheel and with the number of blades or vanes in the wheel.

Lower outlet velocities than those listed in Table 1 may be employed, but care must be exercised when fans of the forward curved type are used to avoid selecting a fan for operation below its useful range. The useful range of the fans of Table 2 extends over the full length of the performance curve.

In exhaust ventilating systems where the air column moves toward the

fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building as when the air column is moving toward the rooms. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Amply large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers require definite increments of the total pressure which a supply fan in a ventilating system must overcome. These static pressures should be considered when selecting the fan characteristics, speed, and power.

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR FORWARD CURVED MULTIBLADE VENTILATING FANS

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
¼	1000–1100	1520–1700
⅜	1000–1100	1760–1900
½	1000–1200	1970–2150
⅝	1100–1300	2225–2450
¾	1200–1400	2480–2700
⅞	1300–1600	2660–2910
1	1500–1800	2820–3120
1¼	1600–1900	3162–3450
1½	1800–2100	3480–3810
1¾	1900–2200	3760–4205
2	2000–2400	4000–4500
2¼	2200–2600	4250–4740
2½	2300–2600	4475–4970
3	2500–2800	4900–5365

Fans picked within the limits of Table 1 will operate close to the point of maximum efficiency. No attempt has been made to select these limits for quiet operation, since this is a relative term and varies with the type and location of the installation.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where noise prevention is a factor the fan and its driver should have floating foundations.

Fans for Drying

Both axial flow and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing axial flow fans are widely used in the drying field. In drying, these fans may be used with unit heaters where not too much duct work is required and where air is to be delivered against

pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work.

Centrifugal fans of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes of air against all pressures likely to be encountered.

Belt driven fans usually are to be preferred to direct-connected fans since efficient motor speeds do not usually coincide with efficient fan speeds. Replacement of a standard motor is quick and easy if it is belted.

Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers. A throttled outlet reduces the volume and reduces the power.

Due to the low speeds of forward curved multiblade or paddle-wheel type fans, these can be direct-connected to reciprocating steam engines,

TABLE 2. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS WITH BACKWARD TIPPED AND DOUBLE CURVED BLADES

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
1/4	800–1100	2600–3100
3/8	800–1150	3000–3500
1/2	900–1300	3400–4000
5/8	1000–1500	3800–4500
3/4	1100–1650	4200–5000
7/8	1200–1750	4500–5300
1	1200–1900	4800–5750
1 1/4	1300–2100	5300–6350
1 1/2	1400–2300	5750–6950
1 3/4	1500–2500	6200–7550
2	1600–2700	6650–8050
2 1/4	1700–2800	7050–8550
2 1/2	1800–2950	7450–9000
3	2000–3200	8200–9850

and the exhaust steam from the engines may be used in the heating apparatus. In selecting engine driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a dryer varies, and where several fans must operate in parallel, fans are to be preferred which have a continuously rising pressure characteristic, such as is given by backward-curved or double-curved blades. This type of fan is well adapted for direct-connected motors of the higher speeds.

Fans for Dust Collecting and Conveying

The application of fans for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 21. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained

resistance or total static pressure at which the fan is to operate. The selection of a proper size fan is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN CONTROL

Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity or by interposing resistance, as by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it may be exceedingly desirable to reduce the speed at times, while on the other hand, any fan which has its normal speed reduced as much as 50 per cent without *change in resistance* will move only 50 per cent of the air.

DESIGNATION OF FANS

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive.)[3]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word "hand," but specify "clockwise" or "counter-clockwise."

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

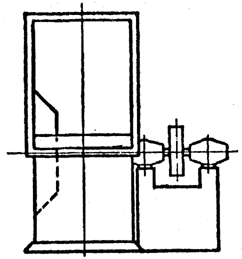
All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

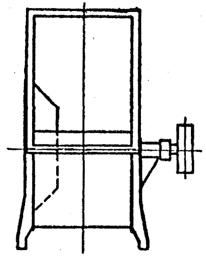
In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 5 are suggested.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a *forward curved blade* type of fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half.

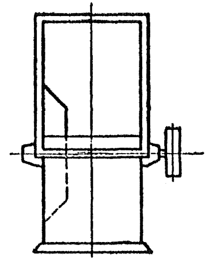
[3]Recommendations adopted by the *National Association of Fan Manufacturers.*



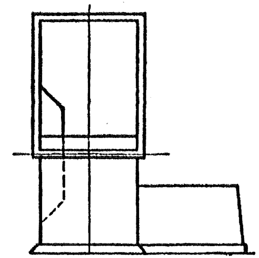
Arr. 1.
For belt drive.
Wheel overhung. Bearings on pedestal.



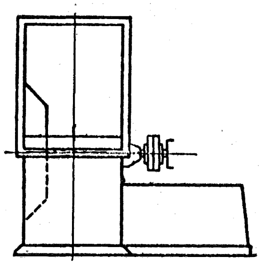
Arr. 2.
For belt drive.
Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.



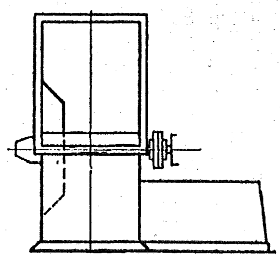
Arr. 3.
For belt drive.
Pulley overhung. Bearings supported on fan housing.



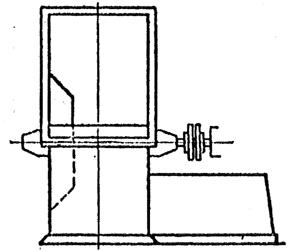
Arr. 4.
For direct drive.
Wheel overhung. No bearings on fan. Wheel mounted on motor or engine shaft. Pedestal for motor or engine.



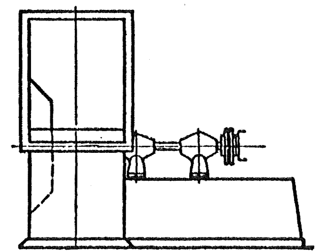
Arr. 5.
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.
Similar to Arr. 6, but with two bearings on fan, and flexible instead of rigid coupling.



Arr. 8.
For direct drive.
Similar to Arr. 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling.

FIG. 5. ARRANGEMENT OF FAN DRIVES

The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan or, having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved* blade types are used, motors with ratings very close to that of the fan horsepower demand can be employed.

Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.*

Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines, and turbines, but as previously stated the drive commonly used is the electric motor.

ELECTRIC POWER

Each type of electric motor and kind of electric current has its advantages and disadvantages as applied to a fan. For motor specifications and standards, the Motor and Generator Standards of the *National Electrical Manufacturers Association* should be consulted.

Direct-connected electric motors usually are very efficient for fan driving because there is no slippage due to belts, and no wear or noise due to chains or gears. There is less maintenance and upkeep to a direct-connected unit, and with an overhung fan wheel on the motor shaft, the usual fan bearings are eliminated.

The disadvantage of a slow-speed direct-connected motor is that it may be unduly large and heavy as well as costly, but this may be offset by the compactness of the unit as a whole due to limited space for fan equipment.

Should anything go wrong with a slow-speed direct-connected motor there may be a considerable delay in securing replacements, as these motors are not usually carried in stock, as is the case with moderately high-speed motors.

If a change of speed is found necessary with a direct-connected motor, it will mean a change of motor, which may necessitate a change in the motor foundation usually built with the fan in such cases. On the other hand, non-direct-connected motors have transmissions subject to wear and slippage, and chains or gears may be noisy with this latter type.

TABLE 3. CLASSIFICATION OF MOTORS

GROUP	SUB-DIV.	TYPE	CURRENT	SPEED CHARACTERISTICS	STARTING TORQUE	STARTING CURRENT	APPLICATIONS
A	1	Shunt wound	d-c	Constant	Medium	High	Fans
	2	Squirrel-cage	a-c	Constant	Medium	High — about six times full load	Fans, centrifugal pumps
	3	Synchronous	a-c	Constant	Medium	Starts as squirrel cage motor	Motor generator sets, air compressors, fans
	4	Slip ring or wound rotor	a-c	Constant	Heavy	Low	Vacuum pumps, air compressors
	5	Double squirrel-cage	a-c	Constant	Heavy	Medium	Frequent and heavy starting loads, pumps, compressors
	6	Low-torque capacitor	a-c	Constant	Light	Low	Direct-connected fans
	7	High-torque capacitor	a-c	Constant	Medium	Low	Belt drive of fans
	8	High-torque capacitor	a-c	Constant	High	Medium	For heavy starting load such as larger fans, pumps, compressors
	9	Repulsion-induction	a-c	Constant	High	Medium	Fans, pumps, compressors
B	1	Brush shifting	a-c	Adjustable	Medium	Low	Stokers, boiler fans
	2	Cumulative comp'd with shunt predominance	d-c	Adjustable	Heavy	High	Pumps
	3	Squirrel-cage, poles can be regrouped	a-c	Multi-speed	Medium	High	Fans, ice machines
C	1	Series	d-c	Variable	Heavy	Low	Fans
	2	Cumulative comp'd with series predominance	d-c	Variable	Heavy	Low	Single-acting reciprocating pumps
	3	Slip ring—using external resistance in secondary	a-c	Variable	Heavy	Low	Fans

However, should a change in speed be necessary where the motor is not direct-connected, changes in speed ratio can easily be accomplished by changing pulleys, sprockets or gears on either the fan or the motor. In the case of a motor breakdown a standard stock motor may easily be substituted.

A type of drive using a wedge-shaped rope-like belt, singly or in multiple, and capable of use on short pulley-centers is very popular, as it enables the use of high speed motors with slow speed fans. The compactness secured by this equipment compares favorably with that of a direct connected layout. This type of drive also is very quiet in operation, being similar to a conventional belt drive in this respect. Alternating current motor designs are such that improved operating characteristics are obtained with the higher motor speeds. Efficiencies and power factors are improved over those in effect with slower speed motors, thus showing a considerable saving in power consumption, and militating in favor of some effective speed-reducing transmission device such as is given by multiple wedge-shaped belts.

Quietness of operation is more readily obtained with moderately high speed induction motors than with low speed motors, as any slight magnetic unbalance in the latter is not as easily heard. Amplifications of motor induction noises in parts of a building remote from the motor equipment sometimes are carried by the steel work, ducts, or piping in the building. There is considerable evidence that these sounds are more easily controlled with high motor speeds than with low ones.

Motors which are practically quiet in operation and free from magnetic disturbing noises can be obtained, and should always be specified for quietness of operation when used for fan installations in buildings where quietness is a factor.

In the construction of fan and motor foundations where the machinery is mounted on the floor or upon a concrete platform, it is a usual practice to install a layer of cork on top of which is laid or floated the base which carries the apparatus. It is essential that the bolts or lag screws which fasten the machines to this foundation shall not extend through to the floor. It is wise to fasten curbs to the floor, these presenting insulated surfaces to the machinery foundation and so preventing it from traveling. Rubber, especially in shear or in tension, is valuable as a sound absorber in foundations for machinery. Steel shoes for fans and motors with rubber inserts are available. Steel springs are also used effectively for this purpose.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 3.

Control for Electric Motors

Very small direct current motors may be started by throwing them directly on the line through a suitable starting switch. The larger sizes require some type of starting rheostat. When speed adjustment is desired, the controller for adjusting the speeds of the motor usually functions also as a starting device.

Alternating current motors of 5 hp and under usually may be thrown directly on the line. It is good practice to use a starting switch equipped

with a thermal overload or inverse time limit overload device. This type of switch provides protection to the motor beyond that given by fuses. Fuses, when used, necessarily must be large enough to take care of the inrush current but this makes them inadequate for protecting the motor under operating conditions. The thermal overload device allows for this inrush and does not function until an overload has become persistent, the time element depending upon the percentage of overload beyond the rating of the element. This type of switch is available for manual operation and also is furnished in the magnetic type for remote operation by push button, or for operation by other types of pilots, such as pressure switches and thermostats.

On standard squirrel cage motors above 5 hp a starting compensator usually is employed to keep the inrush current within the limits specified by the local power companies. Compensators may be obtained in transformer types and primary resistor types, and usually are furnished for manual operation. They can be secured for remote control also, but necessarily are expensive. However, the new type of high reactance, self-starting motors usually may be thrown across the line up to 30 hp in size, and still have their inrush current within the limits of the rules of the *National Electric Light Association*. With this type of motor a magnetic contactor usually is used. This device may be operated from a remote point by push button, if desired. These magnetic contactors are furnished usually with thermal overload and no-voltage protection.

For remote operation of motors through magnetic starters, the operating buttons may be located in the engineer's or manager's office, and tell-tale indicating lamps may be wired up with the circuit to indicate whether or not the unit is in operation. This type of control is very desirable in large buildings where the engineer is to have complete charge of the ventilating system.

Remote or automatic control of the units may be effected also by pneumatic or hydraulic apparatus, or by thermostats or by pressure devices which are provided with electric contacts for starting or stopping the units upon reaching certain conditions.

Variable speed slip ring motors and direct current motors may also be arranged for remote speed control by means of pre-set automatic regulators, where the operating speed of the motor is set by a dial-switch (which may be near the fan or at a remote point) and the motor is then automatically controlled at any given speed merely by operating the remote control push button for starting or stopping the equipment.

Arrangements may be made for remote control of fan motors, or for automatic control by influence of temperature. Remote control may be by pneumatic or by hydraulic manipulation as well as by electrical means.

In many large ventilating systems which have heating plants in connection, steam engines are used to operate fans. A medium speed steam engine, exhausting at low pressure into the heating system is a very economical source of power, is quiet in operation, and has a wide range of speed variation. The steam economy of such an engine usually is of little importance, since the engine serves as an auxiliary to the pressure-reducing valve interposed in such cases between the boiler and the heaters.

Internal combustion engines and line shafting often are used for fan

driving, requiring clutches or shift-belts with loose pulleys in order to secure proper starting and control.

Ability to adjust the speed of ventilating fans is desirable as a measure of economy and adaptability to varying loads, but where such adjustments are provided very definite speed and pressure indications should be supplied at the controller, since without them in most cases the operator would be compelled to guess at the output.

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PROBLEMS IN PRACTICE

1 ● In a public building, what type of fan is suitable for:

a. A supply fan?

b. An exhaust fan?

a. The centrifugal housed fan is well suited for this work. The various types are the forward curved blade, the radial blade, the full backward curved blade, and the medium speed double curved blade with backward tip. When direct connected motors are to be used, the backward tip fans, on account of their speeds, are better adapted. This type has the added advantage of having a limiting horsepower characteristic which will prevent an overload on the motor. Where the belt drive is used, all of the above types are suitable.

b. For exhaust work all of the above types, as well as disc and propeller fans are suitable, although the latter are seldom used except where there is little or no duct work connected to the fan.

2 ● In selecting fans for quiet operation in public buildings:

a. Should the outlet velocity of the fan be limited?

b. Should the tip speed of the fan be limited?

a. Because all commercial fans operating at pressures suitable for this class of work would be considered noisy if the fan were to discharge directly into the room, and because the duct system on the fan discharge is depended upon to absorb a reasonable amount of fan noise, it is desirable to have a moderate run of duct work with some bends and elbows included as sound deadeners. Where this duct is of necessity very short, the outlet velocity must be kept down to the lower limits recommended in this chapter or else an efficient sound absorber must be used. The experience of the engineer must be his guide in determining the allowable outlet velocity in each individual case.

b. Tip speed should not ordinarily be limited, because different types of fan blades have entirely different allowable tip speeds for quiet operation. A fan having a backward blade at the tip can run at much higher tip speed than can a forward curved or a straight blade fan, with the same degree of quietness.

3 ● Is a direct connected or a belted fan preferable in public building work?

Where space is at a premium, direct connection is best. Next in space economy is the short V-belt drive. The flat belt drive fan requires the greatest floor space. In this class of work, pressures are usually so low that even with the high speed fans the motor cost is greater for direct connected units than for belt drive fans.

4 ● a. What type fans are used in industrial work?
b. What outlet velocity is suitable?

a. All of the centrifugal types are suitable; the disc and propeller types are suitable for low pressure work, or they are often used as exhausters.

b. The outlet velocities on fans for industrial work can be much higher than can those in public building work, where quietness is essential. Fans should be selected with outlet velocities as recommended in this chapter, using the upper limit of velocities.

5 ● Are direct connected or belted fans preferred in industrial work?

In industrial applications, fans are often advantageously direct connected to motors. The pressures are usually high enough to use standard motor speeds. The high speed types of fans have limiting horsepower characteristics so that little margin in power must be provided in the driving motor. Belted fans may be used, but where high power is required a special arrangement is often necessary for shaft and bearings on account of the weight of the sheave and the belt pull.

6 ● A forward curved multiblade fan which requires 5.4 bhp is delivering 22,800 cfm at 70 F against a resistance pressure of 1 in. of water at an outlet velocity of 1440 fpm:

a. What is the static efficiency?
b. What is the total efficiency?

a. 66.3 per cent (see Equation 1).

b. 74.5 per cent (see Equation 2).

7 ● If the above fan has a 54-in. diameter wheel and operates at 193 rpm, will it be suitable for a ventilating installation where a minimum of noise is desirable?

Yes. The tip speed will be 2720 fpm and this, together with the 1440 fpm outlet velocity, falls within the limits given in Table 1 for 1-in. resistance pressure.

8 ● Assuming that a 7½ hp constant speed, high reactance type, self-starting electric motor is used to drive the above fan, what electrical starting apparatus should be used for control from a remote point?

An across-the-line type magnetic push button starter with indicating lamps to show whether or not the unit is in operation.

9 ● What objectionable feature is inherent in the ordinary propeller fan when it is operating at high resistance pressures?

It must operate at a high speed with consequent noise.

10 ● At what point should a fan be selected for operation, and why?

At its point of maximum efficiency because the cost of operation and the noise produced will be least.

Chapter 18

SOUND CONTROL

Measurement of Noise, Noise in Buildings, Coefficients of Absorption, Insulation of Air-Borne Sound, Location and Insulation of Equipment Room, Insulation of Machinery and Solid-Borne Vibration, Control of Noise Transmission Through Ducts, Effect of Humidity upon Acoustics

THE ventilating and air conditioning of any space affect its acoustics and become apparent when consideration is given to the requirements for good hearing in any architectural interior. The requirements which must be given careful study are:

1. The room should be free from noise, whether of inside or outside origin.
2. The useful sound, whether speech or music, should be sufficiently loud (with reference to any residual noise) to be heard easily and distinctly.
3. The useful sound should be distributed uniformly in all parts of the room, and the sound reaching the listeners should be free from long-delayed reflections which produce interference or echoes.
4. The room should be free from pronounced resonant tones which may result from either volume or panel resonance.
5. The room should contain sound-absorptive materials in such amounts, and of such qualities, as will provide a proper balance between the persistence and cessation of the articulated components of sound, that is, the reverberation in the room should be long enough to sustain harmony and impart tonal blending to music, and at the same time it must be short enough to prevent the overlapping and confusing of the separate sounds of speech.

Obviously, the first of these requirements is the one which imposes restrictions on the installation of air conditioning or ventilating equipment—the equipment noises must be unobjectionable in occupied rooms—although the fifth requirement is not entirely independent of the humidity and temperature of the air.

LOUDNESS

Loudness is the sensation of sound intensity. When it is said that one sound is louder than another a difference in intensity level is implied. Two identical whistles when sounded together do not make a sound twice as loud as one. It may take ten to make a sound 20 per cent louder than one. It has been found that loudness bears a logarithmic relationship to intensity of sound. On this basis a scale of loudness has been built and a unit, the decibel (db), has been established. This scale is illustrated in Fig. 1 which shows the loudness of some typical noises. The formula for relating loudness and intensity is:

$$L_1 - L_2 = 10 \log_{10} \frac{I_1}{I_2} \qquad (1)$$

where

L = Loudness in db; I = Intensity.

Thus the two whistles made a noise $10 \log_{10} 2 = 3$ db louder than one whistle and the ten whistles, $10 \log_{10} 10 = 10$ db louder than one. It would take a hundred whistles to make a noise 20 db louder than one and a thousand to make a noise 30 db louder.

MEASUREMENT OF NOISE

Since the chief acoustical problem in the ventilating or air conditioning of a building consists of reducing equipment noise, it is necessary to describe methods for measuring noise. The measurement of noise is a relatively new problem, and although there are several reliable methods, there are as yet no standardized units, scales, or instruments for measuring noise[1]. However, the *decibel* (*db*) described above is widely used in this country and England as the standard unit for noise or sound intensity—a unit of the same size, but called a *phon*, is used in Germany—and the zero level of the scale is a barely audible sound. Since the relation between subjective loudness and sound intensity is dependent upon pitch, it is customary to refer loudness to a single frequency. A 1000-cycle tone is generally accepted as the reference frequency, that is, the loudness of any sound is rated in terms of an equally loud 1000-cycle tone. Thus, a noise of 50 db means that the noise would be judged to be of the same loudness as a 1000-cycle tone which is 50 db above the normal threshold of audibility for the 1000-cycle tone.

As the frequencies decrease below 1000 cycles, the ear becomes less sensitive, until at about 30 cycles sounds are no longer audible regardless of their intensity. Similarly, for higher frequencies, the limit of audibility is reached around 7000 cycles. Thus, at frequencies below 1000 cycles, sounds of the same loudness must have a greater intensity than at 1000 cycles. This is particularly fortunate, as otherwise the low frequency sounds would mask all others.

Noise measurements are usually made by one of three methods. The first is the electrical instrument method, which uses a noise meter usually consisting of a microphone, an amplifier, and a galvanometer. Where such a meter is to measure the loudness of a noise without regard to the frequency distribution, it must contain a weighted network which electrically simulates the varying sensitivity of response of the ear to different frequencies. Where it is desired to analyze the character of the sound, filters which shut out all but certain bands of frequencies are used with the meter. A number of manufacturers make such meters.

The second method consists essentially of varying the intensity of an artificially generated sound until the noise generated is masked by the noise being measured. Obviously, this method is subject to human errors in observation to which the instrumental method is not, but in the hands of

[1]See Proposed Tentative Standards for Noise Measurement, and Proposed American Tentative Standard Acoustical Terminology of the *American Standards Association Sectional Committee on Acoustical Measurements and Terminology*.

Also see How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning*, October, 1931), and Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

a careful observer quite satisfactory results may be obtained. One instrument used is the audiometer, which consists of a buzzer, an ear phone, and a rheostat. The phone is held a fixed distance from the ear while the resistance of the rheostat is varied until the sound of the buzzer, as transmitted electrically to the phone, can no longer be heard. Audiometers are available either for covering all frequencies, as in the noise meter, or for covering certain frequency bands only.

A third method of measuring noise, simple, yet sufficiently accurate for most field measurements, employs only three tuning forks and a stop watch. Forks having frequencies of 128, 512, and 2048 are recommended. The forks must be calibrated. That is, it is necessary to know for each fork (1) the initial intensity, in number of decibels above its threshold, immediately after it has received a standard hit or excitation, and (2) the damping rate, in decibels per second. These calibrations can be made in any well-equipped acoustical laboratory. A *standard hit* or *excitation* can be imparted to the fork by a felt-covered spring hammer, or simply by letting the fork fall from a vertical position through an arc of 90 deg, hitting a suitable pad (such as soft rubber or felt for the 128 and 512 forks and hard rubber for the 2048 fork). The average 512 steel fork will have an initial intensity, when held ¼ in. from the ear with the broad side of the prong facing the ear canal, of about 80 db, and will decay at a rate of about 1.0 db per second. Such a fork will remain audible about 80 sec in a perfectly quiet place, provided the listener has normal hearing. In the presence of a noise, it will remain audible until its tone is just *masked* by the noise. Thus, if a 512 fork, having an initial intensity of 80 db and a damping rate of 1.0 db per second, should be found to remain audible 35 sec in the presence of a certain noise, the masking effect of the noise is 80 − 35, or 45 db.

Procedure

The method of measuring any noise is as follows: The observer, in the presence of the noise, strikes the 128 fork a standard blow. At the same instant he starts a stop watch. The fork is then held in front of the ear canal, and moved back and forth slightly, until the tone of the fork is just completely masked by the noise, at which instant the watch is stopped. This measurement is repeated at least two times. The average time is subtracted from the time the 128 fork remains audible in a quiet place. This difference multiplied by the damping rate of the fork gives the masking effect of the noise at 128 cycles. Similar measurements are made with the 512 and 2048 forks. Measurements of this type give a satisfactory description of both the intensity and the frequency distribution of the noise. The average masking effect of the noise at 128, 512, and 2048 cycles will usually be about 5 to 10 db less than the reading given by a noise meter.

NOISE IN BUILDINGS

Measurements of the intensity of speech, music and noise in many buildings, with special consideration of the noise produced by ventilating equipment, have given the results indicated by Fig. 1. The equivalent loudness of sounds in buildings varies from less than 10 db near the outlet of an air duct in a very quiet sound studio to nearly 100 db in a noisy boiler factory. It will be noted that the noise from the ventilating

fan in a certain high school auditorium was nearly as loud as average speech in a large auditorium. Such an amount of noise is devastating to good acoustics; in fact, it is impossible to hear speech in the presence of such a noise.

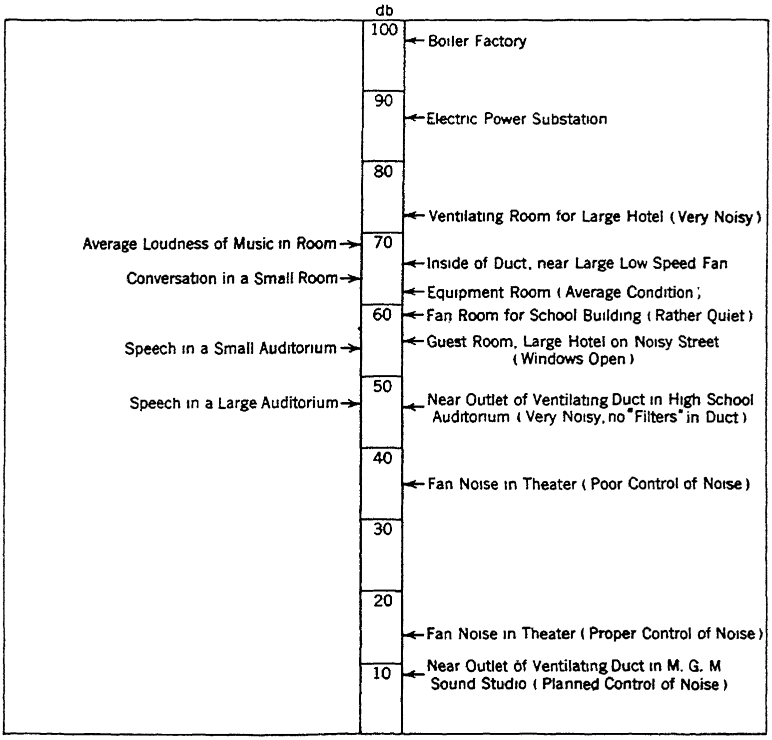

FIG. 1. CHART SHOWING THE EQUIVALENT LOUDNESS (IN DECIBELS) OF SPEECH, MUSIC, AND A NUMBER OF NOISES INCIDENT TO THE VENTILATING OF BUILDINGS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

In every problem of noise reduction in buildings it is necessary to know how much noise can be tolerated. The noise levels given in Table 1 may be regarded as completely inoffensive. They represent what might be termed ideal conditions, not often realized in existing buildings. However, they represent conditions which can be attained by proper control of noise, and the heating and ventilating engineer should aim to provide the degree of quiet specified in the table.

In considering the tolerable room noise level due to heating, ventilating, or air conditioning apparatus, not only must the absolute value of the noise be considered but also its relation to the room noise level without the apparatus running. This is necessary since a large increase of noise subjects the apparatus to serious criticism even though the level may be low. It must also be borne in mind that the noise produced by the ap-

paratus is additive to that of the room without apparatus. Thus if the two are equal, when combined the noise level will be 3 db higher. For these reasons the room noise caused by the apparatus should not exceed the other room noise.

Noise Control

Essential to the design of a satisfactory system are: *first*, a knowledge of the nature and intensity of the noise generated by the various parts of the equipment; *second*, a knowledge of how to vary the noise level between the apparatus and the conditioned room if need be; *third*, a knowledge of the acceptable level of apparatus noise in the conditioned room. Besides these, the engineer must be able to deal with other noises which might enter the room when openings are made into it, such as *cross talk* between rooms connected with common ducts, and noise

TABLE 1. ACCEPTABLE NOISE LEVELS

Talking picture studios	6 to 8 db
Radio broadcasting studios	8 to 10 db
Hospitals	8 to 12 db
Music studios	10 to 15 db
Apartments, hotels, homes, small private offices	10 to 20 db
Theaters, churches, auditoriums, classrooms, libraries	12 to 24 db
Talking picture theaters, small clothing stores	15 to 25 db
General offices	20 to 30 db
Large public offices, banking rooms, upper stories of department stores, restaurants, barber shops	25 to 35 db
Grocery stores, drug stores	30 to 50 db
Accounting and typewriting offices	35 to 45 db
Main floor of department stores	40 to 50 db

transmitted to portions of duct systems outside the conditioned room and thence to its interior.

The problem of apparatus noise is receiving the study of equipment manufacturers who are aiming at both noise reduction and standardization. Some manufacturers now have noise ratings available for their equipment, while some pass each unit of equipment of certain types through sound tests during the course of manufacture.

The problem of noise reduction from apparatus to room must take into consideration and treat separately the three modes of travel of noise to the room: *first*, from the apparatus through the air to the walls of the room and thence to its interior; *second*, through the building structure to the room; *third*, through ducts or openings to the room. Because the noise entering by each of these three channels is susceptible to quantitative analysis, solutions are available. Along with the transmission of sound through the building structure, the engineer must also consider the transmission of vibration, which may also be objectionable. The solution is not complete, however, until the effect of the noise entering the room on the room noise level is determined.

ROOM NOISE LEVEL, COEFFICIENTS OF ABSORPTION

One of the most effective means of reducing noises in ventilating equipment is accomplished by the proper covering of the interior walls and

ceiling of the equipment room, or the inner walls of the ducts, with sound-absorptive materials. The intensity I of a continuous sound in a room is

$$I = \frac{E}{a} \text{ or } \frac{I^{\prime}S^{\prime}}{a} \qquad (2)$$

where

E = the rate of emission of the noise source = $I^{\prime}S^{\prime}$. (The intensities of noises entering the room times the areas through which they enter.)

a = the total amount of absorption supplied by the boundaries and contents of the room.

= $\alpha_1 S_1 + \alpha_2 S_2 + \alpha_3 S_3 + \ldots\ldots$, *where* S_1, S_2, S_3, ... are the areas of the boundary materials for the room, *and* α_1, α_2, α_3, ... are the corresponding coefficients of absorption.

Hence, by increasing tenfold the absorptivity of the boundaries of a room it is possible to reduce tenfold the average intensity of sound in the room; that is, the intensity level would be reduced 10 db.

Thus it is possible to compute the noise level in the room if the intensity of noises entering the room or generated in it are known.

It will be seen that the noise intensity reduction is dependent upon the amount of sound absorption in the room, and that the first units of absorption are more effective than succeeding units. In general, the room noise level will be from 10 to 20 db lower than the air inlet or outlet noise intensity, the 10 db being in the case of bare rooms having large ventilating or air conditioning openings in relation to their size, and the 20 db in the case of rooms having large amounts of absorptive material with small openings. In some cases, the noise level reduction may run up to as much as 30 db, but then the higher sound intensity adjacent to the openings tends to nullify the effects of the extra reduction. Where these openings are large, the local effect on the noise intensity extends some distance from the opening; for instance, a four-square-feet opening might have a local effect within ten feet, while a one-half-square-foot opening would have a local effect within only five feet.

The coefficients of sound-absorption for a number of standard absorptive materials used, or suitable for use, in equipment rooms are given in Table 2. Coefficients are given for frequencies of 128, 512, and 2048 cycles. Where the frequency of the noise is not known, the values for 512 or 128 cycles are usually used.

INSULATION OF AIR-BORNE SOUND

The transmission of air-borne sounds through rigid partitions is accomplished primarily by the diaphragm-like vibrations of the partition. The weight per square foot of the wall is the determining factor, and the insulation value of a wall, in terms of the transmission loss in decibels, is proportional to the logarithm of the weight per square foot. Other factors, such as size, stiffness, composition, manner of mounting, and the use of multiple structures separated by air spaces or flexible connectors, contribute to the effective insulation. If the coefficients of sound transmission of different types of structures and the noise intensity in the space adjoining a room are known, it is possible to calculate the noise intensity in a room by the use of formula 1 and the following formula:

$$I^{\prime} = I^{\prime\prime}\tau \qquad (3)$$

TABLE 2. COEFFICIENTS OF SOUND ABSORPTION[a]

MATERIAL	THICKNESS (INCHES)	COEFFICIENTS OF SOUND ABSORPTION		
		128 Cycles	512 Cycles	2048 Cycles
Acoustex 60, spray painted	1	0.16	0.51	0.72
Acousti-Celotex, Single *B*	⅝	0.11	0.45	0.68
Acousti-Celotex, Triple *B*	1¼	0.20	0.75	0.67
Acoustic Flexfelt	....	0.27	0.56	0.68
Acoustone	1	....	0.66	0.69
Akoustolith plaster	½	0.21	0.29	0.37
Akoustolith *A*, Tile	1	0.14	0.48	0.83
Brick wall, unpainted	18	0.024	0.031	0.049
Calicel	1	0.23	0.72	0.71
Corkoustic, Type C	1½	0.08	0.61	0.64
Glass	....	0.035	0.027	0.020
Insulite Acoustile, Type 44	1⅜	0.26	0.50	0.61
Kalite, with three coats lacquer	¾	0.35	0.43	0.45
Macoustic Plaster, stippled to depth of ½ in.	½	0.13	0.31	0.58
Masonite	7/16	0.18	0.32	0.33
Plaster, gypsum on hollow tile	....	0.013	0.020	0.040
Plaster, gypsum, scratch and brown coats on metal lath on wood studs	....	0.020	0.040	0.058
Plaster, lime, sand finish, on metal lath	¾	0.038	0.060	0.043
Poured concrete, unpainted	....	0.010	0.016	0.023
Rockoustile	1	0.18	0.57	0.72
Sabinite	½	....	0.34	0.49
Sanacoustic Tile	1¼	0.19	0.79	0.74
Stuccoustic Plaster, Type XB	¾	0.29	0.59	0.72
Transite Tile	1	0.19	0.81	0.72
Trutone Tile	1⅛	0.31	0.57	0.64
Wood sheathing, pine	¾	0.098	0.10	0.082
Wood, varnished	....	0.05	0.03	0.03

[a]*Architectural Acoustics*, by V. O. Knudsen, pp. 219, 220, 240-251.

where

I'' = noise intensity in space adjacent to room.
τ = coefficient of sound transmission.

Coefficients of sound transmission for some common walls are shown in Table 3.

Example 1. Suppose the brick wall between an equipment room and an adjacent auditorium has an area of 200 sq ft and a coefficient of sound of 0.00001 (see Table 3); that the auditorium contains 2000 sabines[2] of absorption; and that the noise level in the equipment room is 70 db above zero level.

$$70 - 0 = 10 \log_{10} \frac{I''}{I_0} \text{ (from Formula 1)}$$

$$\frac{I''}{I_0} = 10^7$$

$$\frac{I'}{I_0} = 10^7 \times 0.00001 = 100 \text{ (from Formula 3)}$$

$$\frac{I}{I_0} = 100 \times \frac{200}{2000} = 10 \text{ (from Formula 2)}$$

[2]A *sabine* is 1 sq ft of totally absorptive surface.

Room loudness $= 10 \log_{10} 10 = 10$ db

If the sound absorption in the auditorium had been as small as 200 sabines, the sound intensity in the auditorium would have been 10 times as great and the noise level in the auditorium would have been 20 db.

If the rest of the auditorium has an area of 20,000 sq ft with a surrounding noise intensity of 50 db ($I'' = 10^5$) the noise level due to all of the noise entering through the wall would be found as follows:

$$\frac{I'}{I_0} = 10^5 \times 0.00001 = 1$$

$$\frac{I}{I_0} = 10 \text{ (Through equipment wall)} + 1 \times \frac{20,000}{2000} = 20$$

Room loudness $\times\ 10 \log_{10} 20 = 13$ db

Now suppose that there is also a duct having 20 sq ft outlet connecting the room with apparatus having a noise level of 70 db ($I'' = 10^7$) and suppose that there is an assumed ttenua tion in the duct equivalent to a transmission factor of 0.0002. Then,

$$\frac{I'}{I_0} = 10^7 \times 0.0002 = 2 \times 10^3$$

$$\frac{I}{I_0} = 20 \text{ (from above)} + 2 \times 10^3 \times \frac{20}{2000} = 40$$

Room loudness $= 10 \log_{10} 40 = 16$ db

It may be seen how the energies of noises entering a room are added to obtain the final room noise intensity.

The average coefficients of sound transmission (128 to 4096 cycles) for a number of walls and of floor and ceiling partitions are listed in Table 3.

TABLE 3. AVERAGE COEFFICIENTS OF SOUND TRANSMISSION FOR BUILDING PARTITIONS[a]

DESCRIPTION OF PARTITION	AVERAGE COEFFICIENT
Brick panel, Mississippi, 8 in.; plastered both sides gypsum brown coat, smooth white finish; good workmanship	0.000010
Brick wall, 2½-in. plaster both sides	0.000032
Brick wall, 2½-in., 2-in. furring strips, ½-in. rigid insulation lath plastered both sides	0.0000016
Brick wall, 4 in., 2-in. furring strips and ½-in. rigid insulation lath, plaster, on one side; other side plastered directly on brick	0.0000040
Concrete flat slab floor construction, reinforced; floating floor consisting of nailing strips, rough and finish flooring; ½-in. rigid insulation furred out and applied as ceiling	0.0000020
Glass, plate ¼-in.	0.0010
Glass, plate ¼-in. double glazed, 1½-in. separation	0.0001
Metal lath, double, on 1½-in. channels, ¾-in. gypsum plaster; without cross bracing clips; 4 in., connected at edges only	0.000016
Tile, hollow clay partition, three cells, 4 in. x 12 in. x 12 in., wood furring strips, ½-in. rigid insulation, gypsum brown coat, smooth white finish	0.0000050
Wood joists, lower side plastered on wood lath; floating floor consisting of nailing strips, rough and finish flooring	0.0000050
Wood studs, four-paper plaster board, three-coat smooth finish gypsum plaster	0.000010
Wood studs, two ½-in. sheets rigid insulation both sides, joints filled, gypsum scratch and brown coats, smooth white finish	0.000013
Wood studs, 2 in. x 4 in., staggered, metal lath, ½-in. gypsum plaster; 7½ in.; connected at edges only	0.000040

[a] *Architectu al Acoustics*, by V. O. Knudsen, pp. 308-322.

LOCATION AND INSULATION OF EQUIPMENT ROOM

The equipment room, if possible, should be located at a considerable distance from all rooms in which quiet is required. If this is not possible, it is necessary to provide a high degree of insulation against the noise which may be transmitted through the walls of the equipment room, and also against the noise which almost certainly will be communicated through the short ducts. (See discussion of Control of Noise Transmission through Ducts, p. 311.) Three wall sections and two floor and ceiling sections which are satisfactory for the wall insulation of the equipment room are shown in Fig. 2. Other partitions, with their sound insulating values, are listed in Table 3. The addition of absorptive materials (such as are described in Table 2) to the inner walls and ceiling of the equipment room will not only increase the insulation through the walls, but will also reduce the intensity of the noise in the room. The equipment room noise intensity may be figured in the same way as that of the conditioned space, taking the equipment as the source of noise. In case the equipment is subject to considerable vibration it is advisable to provide a separate or *floated* floor.

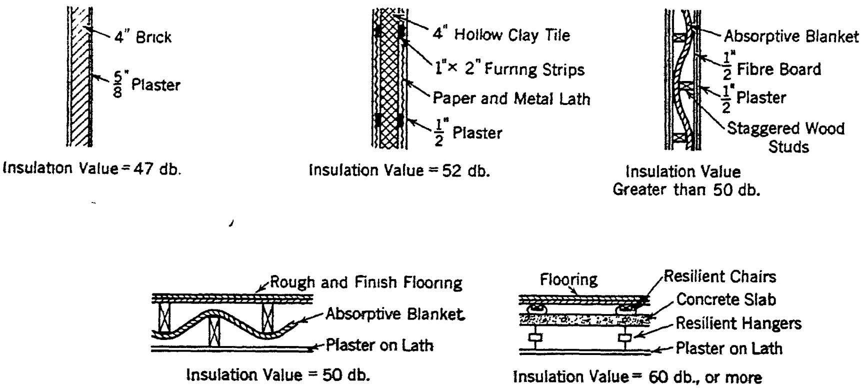

FIG. 2. THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH ARE SUITABLE FOR THE INSULATION OF EQUIPMENT ROOMS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

INSULATION OF MACHINERY AND SOLID-BORNE VIBRATION

Since mechanical vibrations are readily transmitted through the solid structure of a building, it is extremely important in air conditioning that all mechanical equipment in which vibrations are generated be thoroughly insulated from the solid structure of the building. An almost universal notion prevails that the vibrations generated by machinery can be insulated from a building simply by placing a slab of cork or a layer of hair felt between the machinery and the floor of the room. If the machinery is sufficiently heavy, and the cork or felt sufficiently resilient, this expedient may suffice. On the other hand, if the machinery is not suf-

ficiently heavy to *load* the cork or felt support to the extent that the natural frequency of the machinery on the cork or felt is low in comparison with the frequency generated by the equipment, the cork or felt may be of little avail. The insulation of vibration can be accomplished by means of suitable elastic supports or suspensions, but the design of these elastic supports should be based upon calculation rather than guess-work.

The theory of the insulation of vibration was first worked out by Soderberg[3]. If a machine of mass m be supported by an elastic pad the amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by the following equation:

$$\tau' = \sqrt{\frac{r^2 + \frac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi nm - \frac{1}{2\pi nc}\right)^2}} \qquad (4)$$

where

τ' = the so-called *transmissibility* of the support.
c = the compliance (that is, the reciprocal of the force constant).
r = the mechanical resistance owing to the viscous forces within the support.
n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.
m = the mass of the machine to be insulated.

It should be noted that not only must vibrations within the audible range of frequencies be considered, but those in the sub-audible range as well, since these may cause objectionable vibrations. All the possible frequencies should be considered in the calculation. Sometimes beat effects are introduced by slight irregularities of belts or pulleys that have much lower frequencies than those of the rotating elements.

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ' must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 20 vibrations per second.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support

[3]C. R. Soderberg, *The Electric Journal* (January, 1924), and succeeding articles. See also V. O. Knudsen, *Physical Review*, Vol. 32, 1928, p. 324, and A. L. Kimball, Journal *Acoustical Society of America*, Vol 2, 1930, p. 297.

will be worse than nothing. In general, as Equation 4 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building. Furthermore, the machine should rest upon a rigid floor so that the elastic yielding of the floor is prevented from communicating the machinery vibrations to the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approximately because the resistance will be inversely proportional to the thickness and directly proportional to the area of cross-section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 4, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m. For the routine calculations in practice, r may be neglected with only a slight sacrifice of accuracy. Table 4 gives the values of c and r for a number of commonly used flexible materials.

In general, there are two principal points to observe in the design of a flexible support for any piece of equipment, namely, the material should have a relatively large compliance and *it should be loaded to nearly the upper safe limit of loading*. Several flexible metallic supports have recently been developed.

Example 2. A machine weighing 1000 lb has a base area of 20 sq ft. Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most

TABLE 4. COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS[a]

The compliances and resistances given in the table are for specimens 1 in. thick and 1 sq cm in cross-section

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1.10 lb per board foot	12	0.25×10^{-6}	0.15×10^{5}
Corkboard	0.70 lb per board foot	8	0.50×10^{-6}	0.25×10^{5}
Flax-li-num	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^{5}
Celotex	Carpet lining	10	0.40×10^{-6}	....
Celotex	Insulating board	12	0.18×10^{-6}	....
Insulite	Insulating board	15	0.16×10^{-6}	....
Masonite	Insulating board	15	0.12×10^{-6}	....
Anti-Vibro-Block	....	5	0.60×10^{-6}	1.5×10^{5}
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0×10^{-6}	....
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2×10^{-6}	....
Hairfelt	10 lb per cubic foot	1 to 2	1.5×10^{-6}	....

[a]*Architectural Acoustics*, by V. O. Knudsen, p. 278.

machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in. slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than ⅓ lb per square inch. (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^{5}.)

The *transmissibility* is calculated in the following manner:

Mass of machine in grams $= 1000 \times 454 = 4.54 \times 10^{5}$.
Area of base in square centimeters $= 20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^{4}$.

Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^{4}} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of the entire support is $0.15 \times 10^{5} \times 1.86 \times 10^{4} = 0.28 \times 10^{9}$ mechanical ohms (or absolute units). Therefore,

$$\tau' = \sqrt{\frac{(0.28 \times 10^{9})^{2} + \frac{10^{20}}{4\pi^{2} \times 100 \times 0.134}}{(0.28 + 10^{9})^{2} + \left(2\pi \times 100 \times 4.54 \times 10^{5} - \frac{10^{10}}{2\pi \times 100 \times 0.134}\right)^{2}}} = 0.93$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second.

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm. The compliance of the

entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^5 \times 645 = 0.97 \times 10^7$ mechanical ohms (or absolute units). Therefore

$$\tau' = \sqrt{\frac{(0.97 \times 10^7)^2 + \dfrac{10^{18}}{4\pi^2 \times 100 \times 0.39}}{(0.97 \times 10^7)^2 + \left(2\pi \times 100 \times 4.54 \times 10^5 - \dfrac{10^9}{2\pi \times 100 \times 0.39}\right)^2}} = 0.037$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the *transmissibility* is reduced to 0.037, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

CONTROL OF NOISE TRANSMISSION THROUGH DUCTS

The most troublesome sources of noise from ventilating and air conditioning equipment are fan and motor noises which are transmitted through the ducts. The reduction, in decibels, of noise transmitted through a duct, neglecting reflection from ends and bends, is proportional (1) directly to the length of the duct, (2) directly to the perimeter of the duct, (3) inversely to the area of cross-section of the duct, and (4) directly (or at least approximately so) to the coefficient of sound absorption of the material which comprises the interior surface of the duct. It is apparent therefore that long narrow ducts, lined with highly absorptive material, will provide a high degree of insulation against the transmission of noise through ducts. In fact, small ducts (4 in. x 6 in.), made of material having a coefficient of sound-absorption of 0.50, will provide a noise reduction of slightly more than 1 db per linear foot.

As can be seen from an inspection of Table 2, noises of low frequency are difficult to absorb; on the other hand, these frequencies are easily reflected by elbows, branches, and duct ends whereas higher frequencies are little affected. Furthermore, the reflection effects are more pronounced in small ducts than in large ducts. Hence, by introducing into a duct a sufficient length of small, absorptive channels together with a number of elbows or other reflecting elements it is possible to reduce the transmitted noise to any required degree. This applies not only to ducts between the equipment room and other rooms in a building, but also to ducts connecting adjacent or nearly adjacent rooms. By the proper use of such *filters* it is possible to eliminate all of the difficulties which arise in connection with the transmission of sound through ventilating ducts. The problem is an engineering one which can be worked out prior to the installing of the equipment, and it can be calculated in such a way as to meet the most rigorous demands for silent operation. There is a need for quantitative data regarding the attenuation or noise-reduction provided by different types of ducts, but even with the meager data available it is

possible to design filters which will suppress the ordinary noises incident to the ventilating or air conditioning of buildings[4].

In general, the motion of air resulting from the ventilating of rooms is not sufficient to introduce any appreciable difficulty in auditoriums, except where noise may originate from the issuing of high-speed air from nozzles. However, by proper stream-lining of the nozzles, it is possible to work with speeds which are adequate for all practical purposes without producing any disturbing noises. Since sound is propagated with a velocity of more than 1100 fps, the velocity of the air would have to attain speeds of at least 20 to 30 fps before these wind velocities would have any appreciable influence upon the propagation of sound.

If there is to be any appreciable motion of air in an auditorium, it is advantageous to have the upper layers of air moving in a direction from the stage toward the audience, as this will tend to refract the sound waves down toward the audience. However, unless the speed of the air is as great as 20 or 30 fps, the amount of refraction will not be noticeable. Therefore, as a rule the motion of air in an auditorium does not have an appreciable effect upon the acoustical properties of the room.

EFFECT OF HUMIDITY UPON ACOUSTICS

Recent experiments[5] have shown that both the humidity and the temperature of air have a marked influence upon the rate of absorption of high-pitched sounds. Perfectly dry air is less absorptive than air containing any amount of water vapor. At relative humidities of 5 to 25 per cent, the air is highly absorptive but becomes less and less absorptive as the humidity is increased. High-frequency sounds are propagated better in cold humid air than in hot dry air, and since high-frequency sounds are particularly important for the preservation of good quality in speech and music it is advantageous to maintain the air in a room at a relatively high humidity, not less than about 55 to 60 per cent. On the other hand, where it is desirable to absorb all frequency components of sound, as for the reduction of noise in offices, it is advantageous to maintain relatively dry air.

The time of reverberation in a room is given by the following equation:

$$t = \frac{0.049\,V}{4mV - S\log_e(1 - \alpha)} \tag{5}$$

where

V = volume of room in cubic feet.
S = interior surface of room.
α = average coefficient of sound-absorption of the interior surface of the room.
m = the absorption coefficient of the air in the room.

The coefficient m depends upon the frequency of the sound and the humidity (and probably the temperature) of the air. At a temperature of 70 F, and for sound waves having a frequency of 4096 vibrations per second, m = 0.0027 at 25 per cent relative humidity, 0.0018 at 54 per

[4]How Sound is Controlled, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

[5]Effect of Humidity upon the Absorption of Sound in a Room, by V. O. Knudsen (*Journal Acoustical Society of America*, July, 1931). Also see report presented at the May, 1933, meeting of *A. S. of A.*

cent, and 0.0013 at 82 per cent. It will be seen, therefore, that the absorption of sound in the air is twice as great at a relative humidity of 25 per cent as it is at a relative humidity of 82 per cent. This explains why sounds in the open travel so much better on humid days than they do on dry days. Although this dependence of absorption upon humidity is characteristic of low-frequency as well as high-frequency sound, the actual amount of absorption in the air is negligible for frequencies below about 1024 vibrations per second. However, the absorption of the higher frequencies in the air is a significant factor, and its dependence upon humidity calls for careful consideration in planning the air-conditioning equipment for buildings.

PROBLEMS IN PRACTICE

1 ● What are the requirements for good hearing in a room?

Freedom from noise, adequate loudness of speech or music, uniform distribution of sound throughout the room, freedom from echoes and sound foci, no pronounced resonance, and proper reduction of reverberation.

2 ● Why do modern improvements in the acoustics and air conditioning of buildings present new acoustical problems to the heating and ventilating engineer?

In acoustically treated rooms, both outside and inside noise are reduced, and consequently the noise of ventilating equipment becomes more noticeable. The closed windows in air conditioned buildings exclude outside noise, which makes all inside noise from mechanical equipment seem louder.

3 ● Name the acoustical problems which should be solved in connection with the installation of heating or air conditioning equipment.

Selection of quietly operating equipment; adequate insulation of walls surrounding the equipment room; mounting of all vibrating equipment on flexible supports which will eliminate solid-borne vibrations; design of suitable sound filters to reduce the transmission of noise through ventilating ducts; the use of suitably low air speeds and streamlining, where necessary, to prevent *eddy* noises.

4 ● Are good heat insulators also good sound insulators?

As a rule, no. Blankets and felted materials offer considerable insulation for sounds of high frequency, but very little for sounds of low frequency.

5 ● What is the principal consideration in the selection of elastic supports for the insulation of machinery vibration?

The support should be so compliant that the natural frequency of the mass of the machinery on its elastic support will be low in comparison with the vibrational frequencies which are to be insulated.

6 ● What means should be utilized for preventing air-borne noise from the ventilating equipment from being transmitted through the walls, ceiling, or floor of the equipment room?

Treat the interior walls and ceiling of the equipment room with absorptive material; see that all doors and windows to the equipment room fit tightly in their frames; and use wall, and floor and ceiling partitions which have an insulation value of not less than 50 db.

7 ● Name effective methods for reducing the transmission of sound through ventilating ducts.

Line the ducts with sound absorptive material, or use suitable sound filters made up of long channels of small cross-sectional area, lined with sound absorptive material.

8 ● What are the effects of humidity and temperature on the absorption of sound in air?

The absorption increases with a rise in temperature, and decreases for relative humidities above about 20 per cent. A relative humidity of 55 to 60 per cent is advantageous acoustically in large auditoriums.

9 ● How may sound be measured and what are the advantages of the methods available?

Three practical methods are now available to the heating and ventilating engineer, namely:

a. The noise meter method.
b. The audiometer and ear method.
c. The tuning fork and ear method.

Except for instrument adjustments and the use of the eye in reading a meter, the human element does not enter into measurements made with the noise meter, so it is to be preferred, if available. The tuning fork method is relatively cheap and simple and sufficiently accurate for most field work. The audiometer and ear method ranks between these two in preference.

10 ● What are some of the more important sources of noise in buildings, for which the heating and ventilating engineer may be held responsible?

a. Furnace room equipment.
b. Radiators and piping.
c. Uncalked openings in walls around pipes and ducts.
d. Ventilating fans, if noisy in operation and not isolated from the building structure by properly designed vibration damping foundations.
e. High air velocity in ducts.
f. Ventilation fan rooms not insulated acoustically from parts of the building where noise would be objectionable.
g. Ventilating ducts without flexible non-metallic sleeves in them to break metallic sound conducting paths.
h. Cross connection of rooms acoustically through ducts.
i. Ventilating ducts without sound absorbing lining, if required.
j. Unit heaters and ventilators.
k. Unit air conditioners.

11 ● The noise level in the fan room directly under the main floor of a theater is 70 db. The floor is constructed as described in Item 5, Table 3. What is the fan noise level in the theater?

According to Table 3, the average coefficient of sound transmission, t, of such a floor construction is 0.0000020. The transmission loss through the floor, expressed in db, is:

$$\begin{aligned} TL &= 10 \log_{10} \frac{1}{t} \\ &= 10 \log_{10} \frac{1}{0.0000020} \\ &= 57 \end{aligned}$$

The fan noise level in the theater would, therefore, be 70 db less 57 db, or 13 db, which, according to Table 1, is an acceptable level.

Another way of arriving at the same result is by use of Formula 3, in which I' is the in-

tensity of fan noise as measured in the theater, and I'' its intensity as measured in the fan room, I_0 being the reference intensity in both cases, while τ is 0.000002 or 2×10^{-6}.

$$\frac{I''}{I_0} = 10^7$$

$$\frac{I'}{I_0} = 10^7 \times 2 \times 10^{-6} = 20$$

Noise level $= 10 \log_{10} 20 = 13$ db.

12 ● Measurements made separately of the noises from different sources prevailing in a large, noisy banking room revealed the following average noise levels:

- **a. From the street through windows, doors, and walls, 40 db.**
- **b. From adding machines, typewriters, human movements and conversation, 60 db.**
- **c. From the ventilating system, 50 db.**

What was the total noise level of the room?

Calling I_s, I_b, and I_v the intensities of the street, banking room, and ventilation noises, respectively, and I_0 the reference level, we have:

$$\frac{I_s}{I_0} = 10^4 \qquad \frac{I_b}{I_0} = 10^6 \qquad \frac{I_v}{I_0} = 10^5$$

The total intensity, I, will be $I_s + I_b + I_v$

The intensity level is $10 \log_{10} \frac{I}{I_0}$

$$= 10 \log_{10} \frac{(I_s + I_b + I_v)}{I_0}$$
$$= 10 \log_{10} (10^4 + 10^6 + 10^5)$$
$$= 60.4 \text{ db}$$

Note that the total loudness level is not much above the level of the loudest noise. While noise intensities may be added arithmetically, noise levels expressed in decibels cannot be so added.

13 ● A ventilating fan room 30 ft by 30 ft by 12 ft has brick walls, a concrete floor, and a concrete ceiling. How much will the noise level of this room, expressed in decibels, be reduced by applying sound insulating material (coefficient of absorption 0.6 at 512 cycles) to two walls and the ceiling?

Use Formula 2:

$$I = \frac{I'S'}{a} \text{ before applying material}$$

$$I_1 = \frac{I'S'}{a'} \text{ after applying material}$$

$$\frac{I}{I_1} = \frac{\frac{I'S'}{a}}{\frac{I'S'}{a'}} = \frac{a'}{a}$$

Referring to Table 2:

$$a = (4 \times 12 \times 30 \times .031) + (2 \times 30 \times 30 \times .016) = 73.4$$

$$a' = (2 \times 12 \times 30 \times .031) + (30 \times 30 \times .016) + (2 \times 12 \times 30 \times 0.6) + (30 \times 30 \times 0.6) = 1008.7$$

$$\frac{I}{I_1} = \frac{a'}{a} = \frac{1008.7}{73.4} = 13.7$$

Noise level reduction $= 10 \log_{10} \frac{I}{I_1} = 10 \log_{10} 13.7 = 11.4$ db.

14 ● What relation does the movement for the suppression of noise bear to the trend toward air conditioning of offices and other places in cities where people work or congregate?

Very important sources of disturbing sounds are the various street noises that gain entrance, not only through open windows but to a certain extent even through closed windows. If windows are to be kept closed to exclude noise, air conditioning is a practical necessity, especially in summertime. Summertime air conditioning makes use of awnings, which are not only desirable but economical in that they keep down cooling loads. To obviate condensation and frost on windows, wintertime air conditioning calls for storm sash or double glazing which in turn reduces the transmission of street noises

Chapter 19

AIR DISTRIBUTION

Warm Air Systems, Combined Systems, Split Systems, School Buildings, Theaters, Upward System, Downward System, Vanes

TO produce proper air distribution in a room to be ventilated, heated, or cooled by air, the design and location of the air supply inlets and exhaust outlets must be carefully considered. A system may fail though it handles the proper amount of air if such important design principles are ignored.

WARM AIR SYSTEMS

With gravity warm air systems, it has been the practice to place the supply registers in or near the floor of each room and to place the return grille in the floor of the first story. When there is mechanical air circulation, the supply ducts may be extended to the outside walls and the air discharged into the rooms near their cold exposures; on the return side a grille is placed in or near the floor at a central location, or individual return grilles are provided, usually at the corner of the room opposite the supply register.

These arrangements are usually satisfactory for heating (Fig. 1) but not for cooling (Fig. 2). If cool air is introduced at one side of the room at the floor, and if the escape opening for the heated air to be displaced by the cool air is at the floor at the other side, the cool air will travel across the floor and escape through the vent or return air opening, and thus not appreciably affect the warmer air in the upper part of the room.

The air supply opening will serve satisfactorily if located high on an interior wall opposite the exposed wall, and this location answers well also for gravity indirect heating. The corresponding return air arrangements, however, apparently are not subject to exact rules, but must be adapted to circumstances. For example, where the building is compact, with a first story having rooms open to each other, a single, centrally-located return at the floor functions satisfactorily for heating, and if the second story bedrooms are also compactly arranged no individual return from each will be necessary. On the other hand, any room which is unusually exposed, which is especially remote with reference to the other rooms, or which is apt to be tightly closed most of the time, should have a controlled return grille and duct. With a mechanical warm air system, this return may be close to the floor, either below the supply grille or under windows or other cold exposures, and with a gravity system it may be close to the floor at the opposite side of the room from the supply grille.

There is always an advantage in keeping the warm air ducts concen-

trated nearer the furnace and not exposing them to the influence of back drafts of cold air by locating them in outside walls.

COMBINED SYSTEMS

For a combined mechanical heating and cooling system using refrigeration for cooling, no particular change in the ducts usually is necessary. It is desirable from an economic standpoint to take advantage of the natural tendency of the cooler air to remain below the warmer air overhead, and anything which will bring about such stratification will effect an economy in refrigeration.

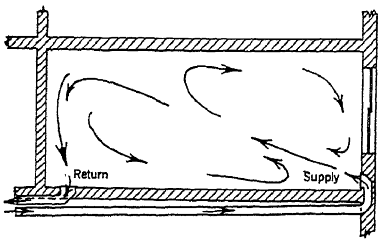

FIG. 1. AIR CIRCULATION WHEN HEATING WITH LOW SUPPLY AND RETURN OPENINGS

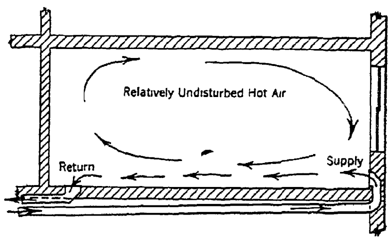

FIG. 2. AIR CIRCULATION WHEN COOLING WITH LOW SUPPLY AND RETURN OPENINGS

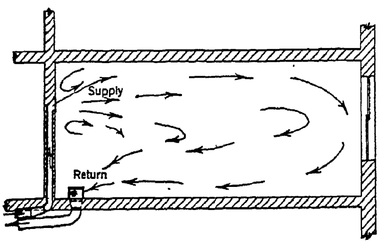

FIG. 3. AIR CIRCULATION WHEN COOLING WITH HIGH SUPPLY OPENING AND LOW RETURN OPENINGS

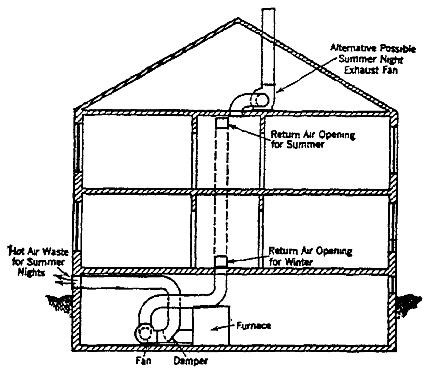

FIG. 4. SECTION THROUGH AN ELEMENTAL MECHANICAL WARM AIR HEATING-COOLING SYSTEM. THE ATTIC FAN IS ALTERNATIVE

If the return ducts of a mechanically operated warm air system are adequate, appreciable cooling may be accompilshed as follows: The fan outlet must have a by-pass leading to a basement window or to a chimney provided for the purpose and the return duct must have an alternative shaft opening into the highest part of the house. At night, in summer, the fan may be operated to exhaust the hot air from the top of the house by the return air duct just described and the fan will blow this heated air out of doors through the window, or preferably, of course, through the chimney. The cooler night air must then enter the house through the

windows, and by its motion and temperature will extract the heat from the walls and furniture. The cost of power for such cooling should be carefully checked against operating with a much smaller volume of air mechanically cooled.

Fig. 3 shows the air circulation when cooling with a high supply opening and a low return opening. The air circulation, when heating, will be substantially the same as when cooling. Fig. 4 shows a section through an elemental mechanical warm air heating-cooling system. The attic plan is alternative. Summer night cooling may, of course, be accomplished by placing an exhaust fan in the attic.

SPLIT SYSTEMS

Many buildings which are heated by radiators or convectors and which have rooms requiring ventilation or cooling have air supply and exhaust systems independent of the radiators or convectors. Such installations are termed *split systems.* When the air enters a room through conventional side wall inlets an occupant may feel comfortable if the air is about the temperature of the room, but the introduction of too cool air may cause a feeling of draft. To correct this draft condition, glass chutes and elaborate diffusers are sometimes provided. The arrangement shown in Fig. 5 for supplying cool air to a room provides satisfactory air circulation in spaces up to 400 sq ft in area with ceilings as low as 8 ft. There is no maximum ceiling limitation as to height.

When the room in question is provided with a unit ventilator which obtains its air supply directly through the wall from out of doors, the distribution with a high velocity air jet passing in an upward direction is quite satisfactory.

The use of unit air conditioners for summer cooling introduces no new features or difficulties which have not already been encountered in winter heating. Conditioners must be provided with positive control by means of valves or dampers, or both, which will prohibit any sudden and wide temperature variation, and keep the entering air not more than approximately 7 deg cooler than the air already in the space. This temperature margin is dependent on various factors including the ceiling height of the room and the velocity of the air at the discharge grille.

SCHOOL BUILDINGS

The air distribution conditions in school building classrooms are not unlike those illustrated in Fig. 1 for mechanical warm air systems and those in Fig. 6 for unit ventilator equipped plants. School rooms which have center-ceiling inlets along the lines of Fig. 5 have given excellent results. It is important that the temperature of the entering air, whether this air be supplied by a local unit ventilator or by a distant central fan, be controlled so that the air cannot enter the room from a side-wall inlet or from a unit ventilator at a temperature more than a very few degrees cooler than that of the air already near the ceiling of the room.

Fig. 7 shows a section through a room equipped with a unit air conditioner or unit cooler. This is typical of the condition in effect when any recirculating room-cooling unit is installed.

Most unit ventilators employ a unique method of air distribution. Its principal feature is that the air is discharged at a high velocity toward the ceiling, with the jet inclined slightly toward the room in order to distribute the air over the ceiling. In designing a unit ventilator installation great pains should be taken that nothing will interfere with the operation of this jet. For this reason unit ventilators should never be installed where there is a beam on the ceiling running at right angles to the direction of the air jet. If ceiling beams cannot be avoided, the unit ventilator should be placed to discharge parallel to the beams.

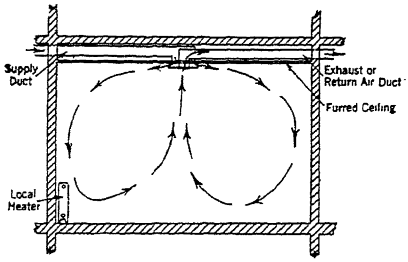

FIG. 5. SECTION THROUGH A RADIATOR-HEATED ROOM

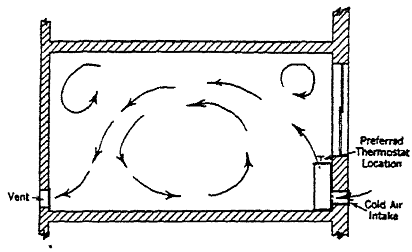

FIG. 6. SECTION THROUGH A UNIT VENTILATOR-EQUIPPED ROOM WHEN HEATING

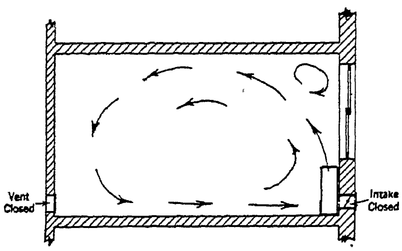

FIG. 7. SECTION THROUGH A UNIT CONDITIONER EQUIPPED ROOM WHEN COOLING

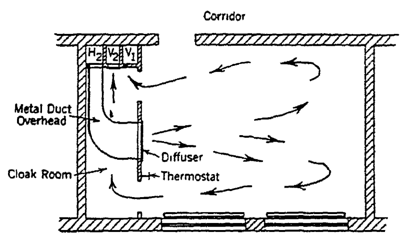

FIG. 8. PLAN OF A CLASSROOM IN A SCHOOL VENTILATED BY A CENTRAL FAN

In Fig. 8 the cloakroom ceiling is furred down so as to conceal the metal air supply duct, which is close to the ceiling. The air for ventilation usually is controlled by a duct thermostat near the fan, at a temperature slightly higher than the temperature required in the room, to allow for heat losses in the duct system.

THEATERS

Theaters are usually ventilated or cooled by introducing preconditioned air. No ventilating system for a theater should be given consideration without definite provision for cooling. Theater cooling generally is far more important than theater heating. There are two widely different methods of theater air distribution, the *downward* and the *upward*.

Downward System

Theaters usually are equipped with downward air distribution with horizontal diffusion of the entering cool air so as to combine it, both as to temperature and dilution, with the heated air which inevitably must rise from the bodies of the patrons. The waste or the recirculated air is withdrawn from the room at the floor. If the theater is large, and if the exhaust openings are placed in the side walls at the floor, drafts may be felt by the people who sit near the openings. There is no objection, however, except that of cost, to the use of small exhaust openings under each seat. These may be cleanable floor grilles or may have mushroom covers.

In a downward system, if the entering cool air is not deflected horizontally, it will fall through the surrounding much hotter air, and will

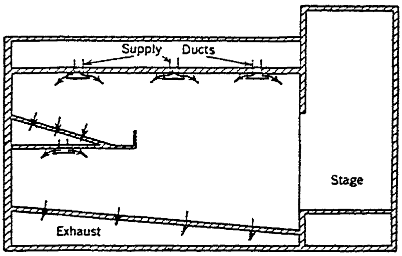

FIG. 9. SECTION THROUGH A THEATER WITH DOWNWARD VENTILATION

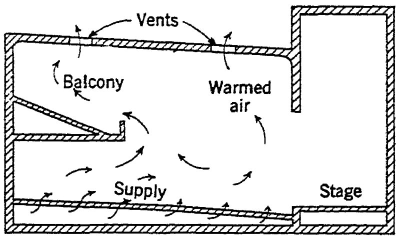

FIG. 10. THEATER WITH UPWARD SYSTEM OF VENTILATION

reach high velocities by the time it strikes the heads of the occupants. Air at a temperature 10 deg below that of the surrounding air is decidedly objectionable when forced over one's head at a velocity of nearly 400 fpm.

Fig. 9 shows a section through a theater with downward ventilation. The deflectors cause the entering cool air to be spread horizontally so that it will mix with the hotter air. The final escape is through well-distributed openings in the floor. There have been cases in which the downward system of air distribution such as that illustrated in Fig. 9 gave trouble due to overheating at the rear, both above and below the balcony, especially when not provided with refrigeration for cooling, and when not adequately controlled. It is especially necessary that adequate removal of the heated air be provided at these low-ceiling points and it is probable that auxiliary exhaust at or through the ceiling after the manner of the arrangements shown in Fig. 5 would be helpful.

Upward System

If no inlet openings are possible in the ceiling, the upward system may be the less objectionable alternative. Fig. 10 shows a section through a theater with the upward system of air distribution. The occupants often suffer from drafts due to the cool air which comes from the unoccupied zones.

When the entire seating area is occupied, the upward system gives little trouble when cooling, and since very little heating is required under such conditions, practically no difficulty is encountered. The maximum

volume of air to be introduced with the upward system is about 25 cfm of air per person at a low velocity, say at 150 fpm (linear), and at a temperature not more than 6 deg below the room temperature. For partial occupancy, higher entering air temperatures can be used with correspondingly less danger from drafts.

VANES

In order to cause the supply air to a room to take a fixed or desired direction when leaving the inlet opening of a flue, stationary vanes may be provided at both the back of the grille and at the grille to direct the air flow. Fig. 11 shows a section through a room inlet opening at the top of a rising flue and indicates the air conditions when no vanes are used. Fig. 12 shows a section through the same room inlet opening when vanes are advantageously placed to direct the flow of air.

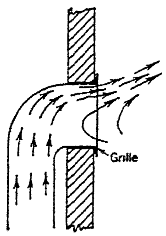

Fig. 11. Air Conditions at Inlet Opening at the Top of a Rising Flue when no Vanes are Used

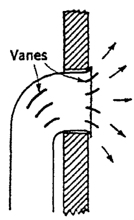

Fig. 12. Air Conditions at Inlet Opening at the Top of a Rising Flue when Directional Vanes are Used

In many theater and commercial installations the ejector-like action of high-velocity air emerging from a duct is taken advantage of, and scientifically proportioned nozzles are installed to cause definite recirculation of the room air.

PROBLEMS IN PRACTICE

1 ● Is the conventional warm air system, employing floor or baseboard supply registers, suitable for heating and cooling?

Floor or baseboard supply registers are suitable for heating service because the natural tendency of warm air is to rise. They are not suitable for cooling because the natural tendency of cool air is to stay near the floor and gradually work its way to the return registers, thus not cooling the air in the upper part of the room. See Figs. 1 and 2.

2 ● What type of air distribution system is suitable for heating and cooling a home?

In order to provide satisfactory cooling without drafts it is necessary to discharge the air at relatively high velocity toward the ceiling from a high point, as shown in Fig. 3. If the register is properly designed and the air capacity is limited to approximately 400 cfm, the cool air will mix with the air in the room before it drops to the occupied zone. However, care must be taken that discharged air does not impinge on beams which would cause the cool air to be deflected downward. This arrangement is also satisfactory for heating.

3 ● Why is the conventional low velocity side wall inlet unsatisfactory for cooling purposes?

With the conventional side wall inlet using velocities of 300 to 400 fpm the discharged air quickly loses its velocity and drops, causing drafts in the occupied zone.

4 ● What method of side wall introduction is satisfactory for cooling purposes with a 12-ft ceiling height?

The method shown in Fig. 3 can satisfactorily circulate air as much as 10 to 15 F below room temperature, provided (1) each jet is limited to 400 cfm, (2) the outlet velocity is high, (3) the air is directed toward the ceiling, and (4) there are no beams on the ceiling. In order to employ this method in a classroom it is usually necessary to have at least three inlets, but even with three inlets the cooling capacity is limited to that obtained by circulating air at 10 to 15 F below room temperature.

5 ● Should unit ventilators be considered as heating units or as cooling units?

Experience has shown that approximately 75 per cent of the time a classroom is occupied the problem is one of cooling rather than one of heating. For this reason unit ventilators should be considered as cooling units.

6 ● What method of air distribution is usually employed with unit ventilators?

Most unit ventilators employ a unique method of air distribution in which the air is discharged at a high velocity toward the ceiling. The air stream is usually inclined toward the room.

7 ● How should a unit ventilator be located in a room that has ceiling beams?

When there are ceiling beams the unit ventilator should be so located that the beams will be parallel with the direction of the air discharge in order that the beams will not deflect the air downward.

8 ● What is the minimum temperature at which unit ventilators can distribute air in a classroom without causing drafts?

Generally speaking, the lowest minimum discharge temperature at which objectionable drafts will not be created is 60 F. Some designers suggest that the discharge temperature can drop as low as 35 F below the room temperature without causing drafts when units are properly installed.

9 ● What is the usual method of ventilating school auditoriums and gymnasiums when unit ventilators are used in the classrooms?

If unit ventilators are used in classrooms the usual method of ventilating the auditorium or gymnasium is to use one or more large units located above and on either side of the stage.

10 ● What is the maximum amount of air which should be discharged from one point in a school auditorium or gymnasium?

The maximum amount of air which should be discharged from one point is 5000 cfm. This limitation applies whether the air is supplied by units or by a central fan from a distant point.

11 ● Are vents required in school classrooms, auditoriums, and gymnasiums?

With both the unit and the central fan systems, vents are usually installed as a certain and positive means of disposing of the vitiated and odoriferous air and also, with the central fan system, for the further purpose of effecting a means of partial recirculation. Natural outward air leakage may take the place of vents, if and when it proves sufficient, but it is usually uncertain, insufficient, and uneconomical. Vents are required by law in some communities. If they are installed, they should be provided with dampers in order that they may be throttled when required and closed at night and during holidays.

12 ● What type of system is generally used in large continuously operated theaters?

Most large continuously operated theaters are provided with complete downward systems of air distribution similar to the one shown in Fig. 9. With this system a large number of inlet openings is provided, each of which discharges air in a thin horizontal stream at high velocity in order that the cool air will be mixed with the air in the theater before it reaches the patrons.

13 ● What system of air distribution is frequently used in smaller theaters?

The system used, particularly where artificial cooling is had, brings air in at high velocity through a large number of small horizontal nozzles located in the rear of the auditorium near the ceiling. This high velocity air mixes with a much larger quantity of air and causes circulation within the theater before it comes into contact with the occupants. With this method care must be exercised not to discharge the air against ceiling beams or projections which may give a downward direction to the cool air before it is thoroughly diluted.

Chapter 20

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Proportioning the Losses, Sizes of Ducts, General Rules, Procedure for Duct Design, Air Velocities, Proportioning the Size for Friction, Main Trunk Ducts with Branches for Public Buildings, Equal Friction Method, Details of Duct Construction

THE flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_V be the velocity head in feet of a fluid, and the velocity, V, be expressed in feet per minute, the fundamental equation is

$$V = 60\sqrt{2g\,H_V}$$

The factor g is the acceleration due to gravity, or 32.16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_V}{\frac{h_V}{12}} = \frac{62.4}{\rho}$$

or

$$H_V = 5.2\,\frac{h_V}{\rho}$$

therefore,

$$V = 1096.5\sqrt{\frac{h_V}{\rho}} \tag{1}$$

where

V = velocity in feet per minute.
h_V = velocity head or pressure in inches of water.
ρ = weight of air in pounds per cubic foot.

For standard air (70 F and 29.92 in. barometer) ρ = 0.07495 lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5\sqrt{\frac{h_V}{0.07495}} = 4005\sqrt{h_V} \tag{2}$$

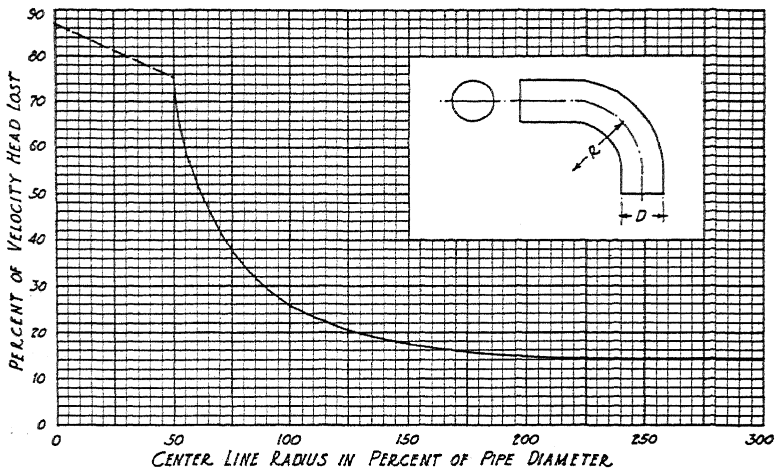

FIG. 1. CURVE SHOWING LOSS OF PRESSURE IN ROUND ELBOWS

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses are those due to the friction of the air against the sides of the duct. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic Losses

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to

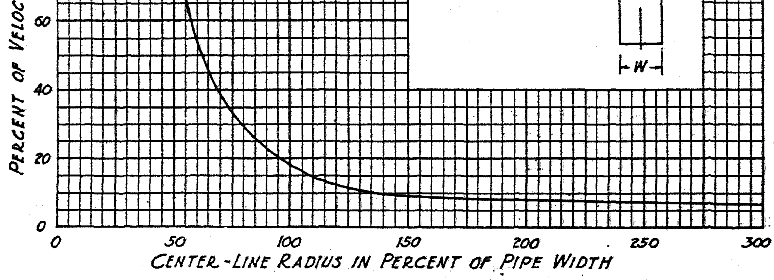

FIG. 2. CURVE SHOWING LOSS OF PRESSURE IN SQUARE ELBOWS

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design. It is customary to express dynamic losses in terms of the percentage of the velocity head; in other words, the percentage of that pressure corresponding to the average velocity in the duct which is expressed in terms of inches of water gage. Figs. 1 and 2 show the effect of changing the radius of elbows of square and rectangular section. These charts are based on tests of pipe elbows of ordinary good sheet metal construction. For example, a five-piece round pipe elbow having a centerline radius of one diameter has a loss of about 25 per cent of the velocity head. At a velocity of 2000 fpm the corresponding head is 0.25 in. water gage, and at this velocity the elbow just referred to would cause a pressure drop of 0.063 in. water gage. Experience has shown that good results may be obtained when the radius to the center of the elbow is 1½ times the pipe diameter. The pressure drop will then be approximately 17 per cent of the velocity head for round ducts, and 9 per cent for square ducts. Very little advantage is gained in making elbows with a radius of more than two diameters.

Friction Losses

Friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter. Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and standard air (70 F and 29.92 in. barometer)

$$h_L = f\frac{L}{D}h_v = \frac{L}{CD}\left(\frac{V}{4005}\right)^2 \tag{3}$$

For rectangular ducts

$$h_L = fL\left(\frac{a+b}{2ab}\right)h_v = \frac{L}{C}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4005}\right)^2 \tag{4}$$

where

h_L = loss of head, inches of water.

$h_v = \left(\frac{V}{4005}\right)^2$ = velocity head, inches of water.

V = velocity of air, feet per minute.

L = length of pipe, D = diameter of pipe, a, b = sides of rectangular duct: all in feet.

f = coefficient of friction.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss.

For all practical purposes C varies only with the nature of the pipe surface: $C = 60$ for perfectly smooth pipe; $= 55$ for pipe as used in planning mill exhaust systems; $= 50$ for heating and ventilating ducts; $= 45$ for

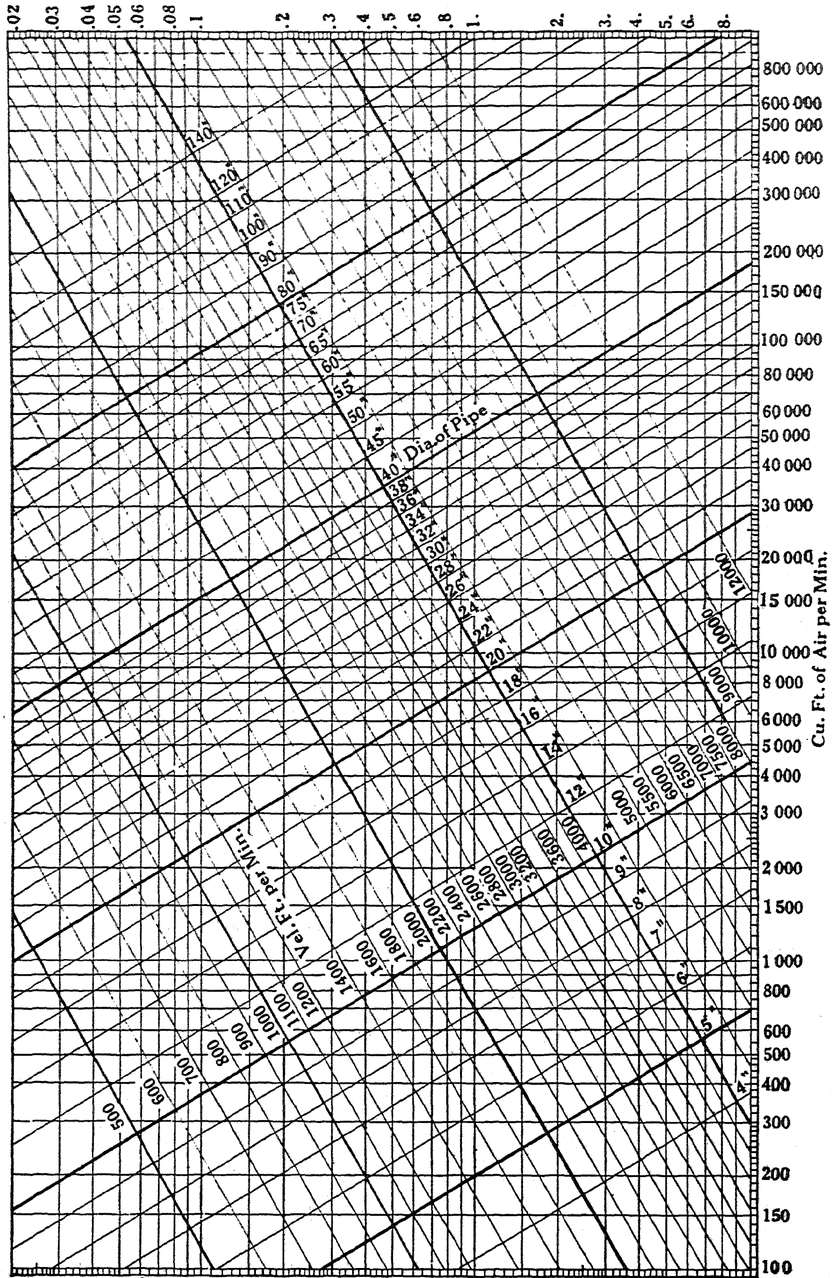

FIG. 3. FRICTION OF AIR IN PIPES

smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzsche states (and numerous tests check very closely) that f varies inversely as the 2/7 power of the pipe diameter, and inversely as the 1/7 power of the velocity, or inversely as the 1/7 power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24-in. galvanized swedged pipe:

$$h_L = 1.1 \frac{L}{CD^{9/7}} \left(\frac{V}{4005}\right)^{13/7} \tag{5}$$

The preceding formulae are based on standard air, and for other conditions the friction varies directly as the air density and inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

Fig. 3 is a convenient chart for determining the friction loss for various air quantities in ducts of different sizes. The general form of this chart is familiar, but it should be noted that it is corrected for changes in the coefficient of friction based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the velocity. Fig. 3 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of 24-in. round galvanized-iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C, the friction loss as read from the chart should be multiplied by $\frac{50}{40}$.

Example 1. Assume that it is desired to pass 10,000 cfm of air through 75 ft of 24-in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 3 and move horizontally left to the diagonal line marked 24-in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.59 in.; then for 75 ft the friction will be $0.75 \times 0.59 = 0.44$ in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Proportioning the Losses

Other losses of pressure occur at the entrance to the duct, through the heating units, and at the air washer. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses ⅓ to ½ and the loss through the heating units at less than ½ of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

DUCT SIZES

The sizes of ducts and flues for gravity or mechanical circulation of air are usually based on the losses due to friction, and these losses must be kept within the available pressure difference. This pressure difference in

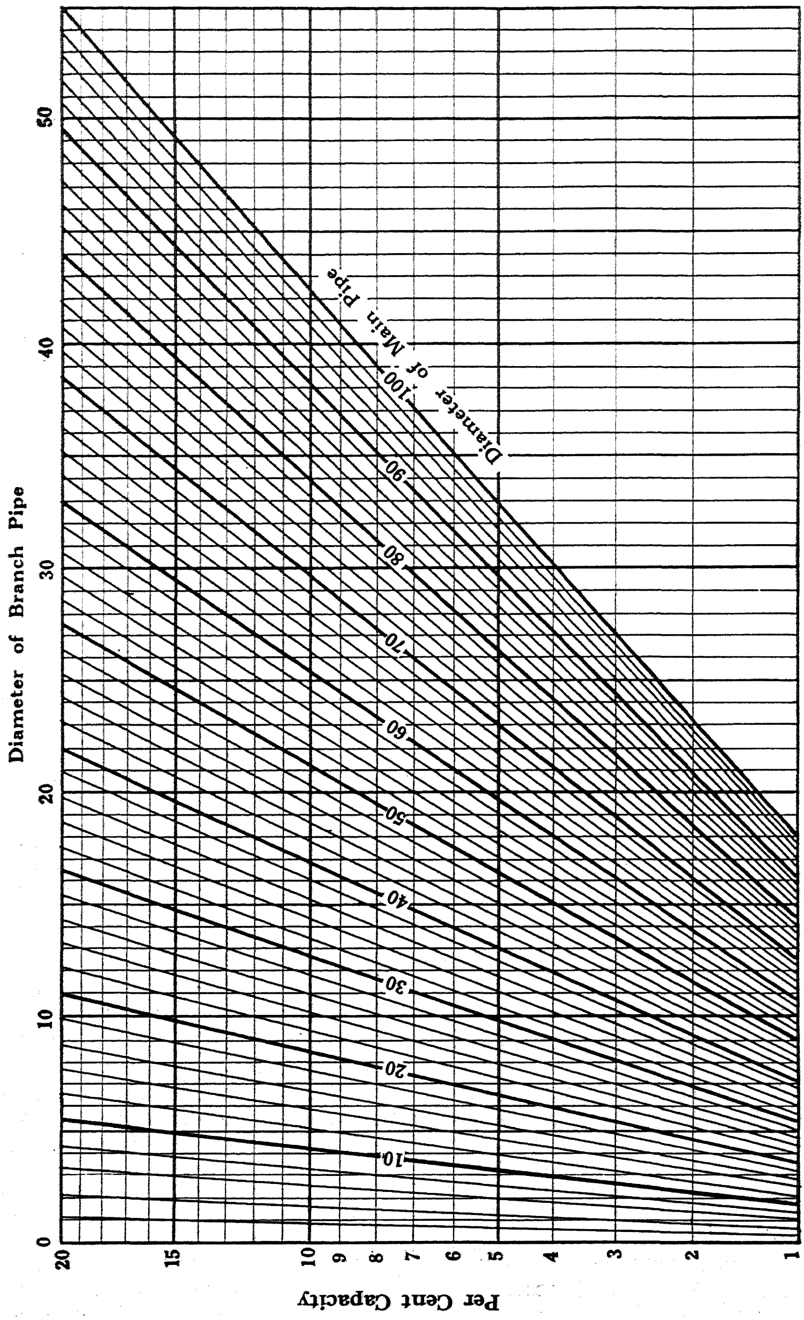

Fig. 4. Main and Branch Pipes for Equal Friction per Foot of Length
(1 to 20 Per Cent Capacity)

mechanical ventilation is that derived from the fan, while in gravity ventilation the aspirating effect due to the temperature and height of the column of heated air causes the pressure difference.

General Rules

The general rules to be followed in the design of a duct system are:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.
2. Sharp elbows and bends should be avoided.
3. The sides of all ducts or flues should be as nearly equal as possible. (In no case should the ratio between long and short sides be greater than 10 to 1.)

Procedure for Duct Design

The general procedure for designing a duct system is as follows:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of heat.
3. Divide the building into zones and proportion the volume of air necessary to supply the heat for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:

    a. Velocity Method. Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections.

    b. Friction Pressure Loss Method. Proportion the duct for equal friction pressure loss per foot of length.

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The following velocities of air are considered standard for public buildings:

1. Through the outside air intakes, 1000 fpm.
2. Through connections to and from heating unit, 1000 to 1200 fpm.
3. Through the main discharge duct, from 1200 to 1600 fpm.
4. In branch ducts, 600 to 1000, and in vertical flues, 400 to 800 fpm.
5. In registers or grilles, 200 to 400 fpm depending upon the size and location. If diffusers of proper design are used, 25 per cent higher air velocities are permissible.

These duct velocities may safely be increased 20 per cent if first-class construction is used to prevent any breathing, buckling, or vibration. High velocities at one point in the system neutralize the effect of proper design at all other points; hence the importance of splitters in elbows and similar precautions. For industrial buildings noise is seldom considered, and main duct velocities as high as 2800 or 3000 fpm may be used where conditions will permit. For department stores and similar buildings,

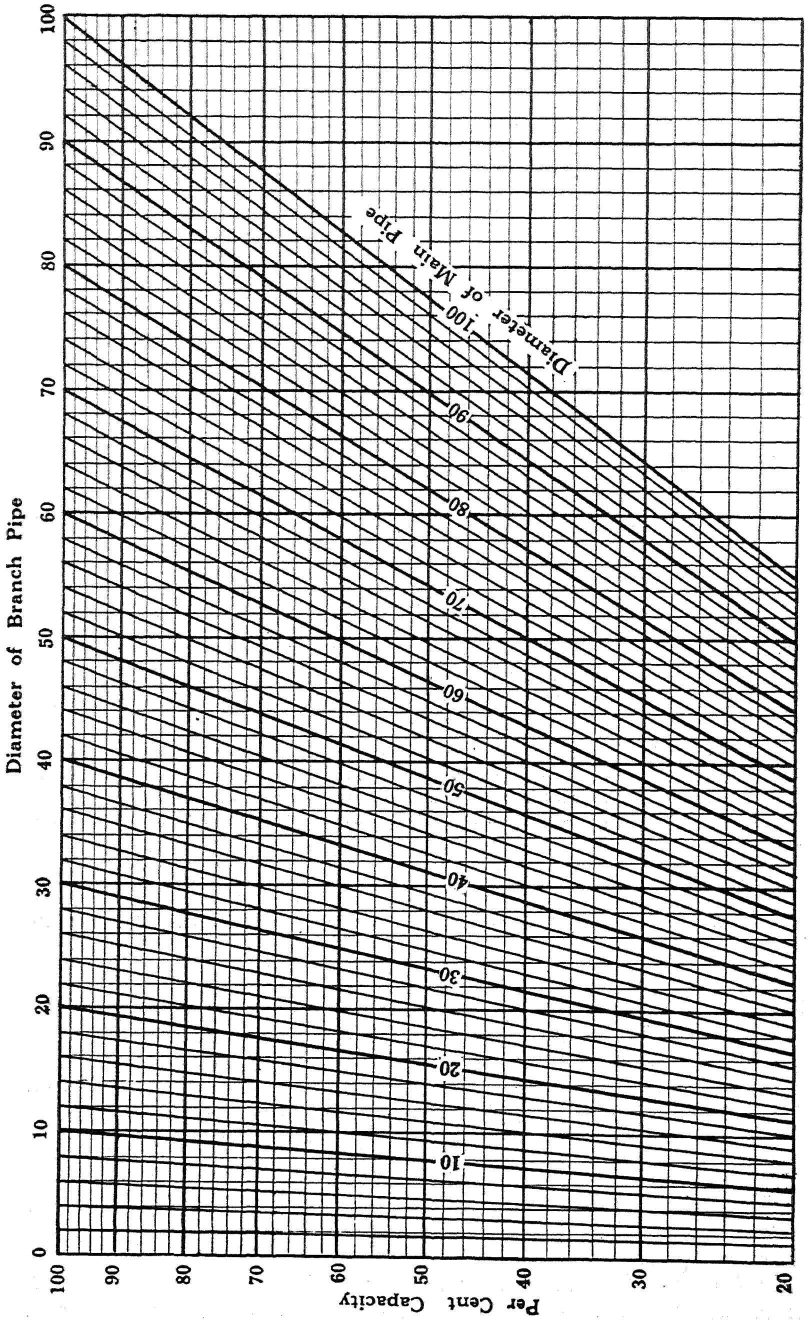

Fig. 5. Main and Branch Pipes for Equal Friction per Foot of Length (20 to 100 Per Cent Capacity)

maximum velocities with good construction and design may be as high as 2000 or 2200 fpm in main ducts, with suitable reduction in branches and outlets. With these velocities first-class duct construction is essential.

Proportioning the Size for Friction

By means of Figs. 4 and 5 the diameter of branch pipes necessary to carry a given percentage of the total air in the main pipe and to maintain equal friction per foot of the length through the entire system may be determined. These charts, as well as Fig. 3, are based on the assumption that the coefficient of friction varies inversely as the 1/7 power of the capacity.

Example 2. Suppose a 60-in. main pipe is to be used, and it is desired to know the size of branch pipe required to carry 50 per cent of the total air in the main. Find 50 per cent at the left of the chart, move right to the 60-in. diagonal line and note directly above at the top of the chart that the branch pipe will be 46.5 in. in diameter.

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalent of rectangular ducts for equal friction and capacity. To obtain the size of rectangular ducts for different capacities, but of the same friction per foot of length, first obtain the equivalent round pipe for equal friction. Thus, if a branch of sufficient size to carry 30 per cent of a 12 x 36-in. pipe is desired, it is found from Table 1 that the main is equivalent to a 22.2-in. diameter round pipe. From Fig. 5, 30 per cent of this is a pipe 14.3 in. in diameter, and referring again to Table 1, the rectangular equivalent branch is a 12 x 14-in., 10 x 17¼-in., or any other desirable combination.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24-in. duct is required, it will be just twice that of a 40 x 12-in. duct, or $2 \times 23.3 =$ 46.6 in.

DUCTS FOR PUBLIC BUILDINGS

A main duct with branches is generally used to convey tempered air for ventilation purposes only. In place of individual ducts, a comparatively large main duct supplies air by branches to the room or rooms. The velocities vary according to the nature of the installation and the degree of quietness required. At the start of the run a velocity as high as 2000 fpm may be used, but this is considered the maximum for public building work, and is reduced to from 400 to 800 fpm in the risers. This duct system may be designed so that the loss of pressure in the branches is equalized in a manner similar to that previously described.

Equal Friction Method

Example 3. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices.

The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-minute air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The clear area of the fresh air inlet is based on a velocity of 1000 fpm or $\frac{22,935}{1000} =$

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION[a]

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	9.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

[a]Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3; 5 × 7 = 6.5.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Continued)

SIDE RECTANGULAR DUCT	26	28	30	32	34	36	38	40	42	44	46	48	SIDE RECTANGULAR DUCT	50	54	60	66	72	78	84	88
26	28.6												50	55.0							
28	29.7	30.8											52	56.1							
30	30.7	31.9	33.0										54	57.2	59.4						
32	31.7	32.9	34.1	35.2									56	58.3	60.5						
34	32.7	33.9	35.1	36.3	37.4								58	59.3	61.6						
36	33.7	34.9	36.1	37.3	38.5	39.6							60	60.3	62.7	66.0					
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8						62	61.3	63.7	67.1					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0					64	62.2	64.7	68.2					
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2				66	63.2	65.7	69.3	72.6				
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4			68	64.1	66.6	70.3	73.7				
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6		70	65.0	67.6	71.3	74.8				
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8	72	65.9	68.5	72.3	75.9	79.2			
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0	74	66.8	69.4	73.3	76.9	80.3			
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0	76	67.6	70.3	74.2	77.9	81.4			
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0	78	68.4	71.2	75.2	78.9	82.5	85.8		
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0	80	69.2	72.1	76.1	79.9	83.6	86.9		
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0	82	70.1	73.0	77.1	80.9	84.6	88.0		
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9	84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7	86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6	88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6	90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6	92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5	94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5	96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

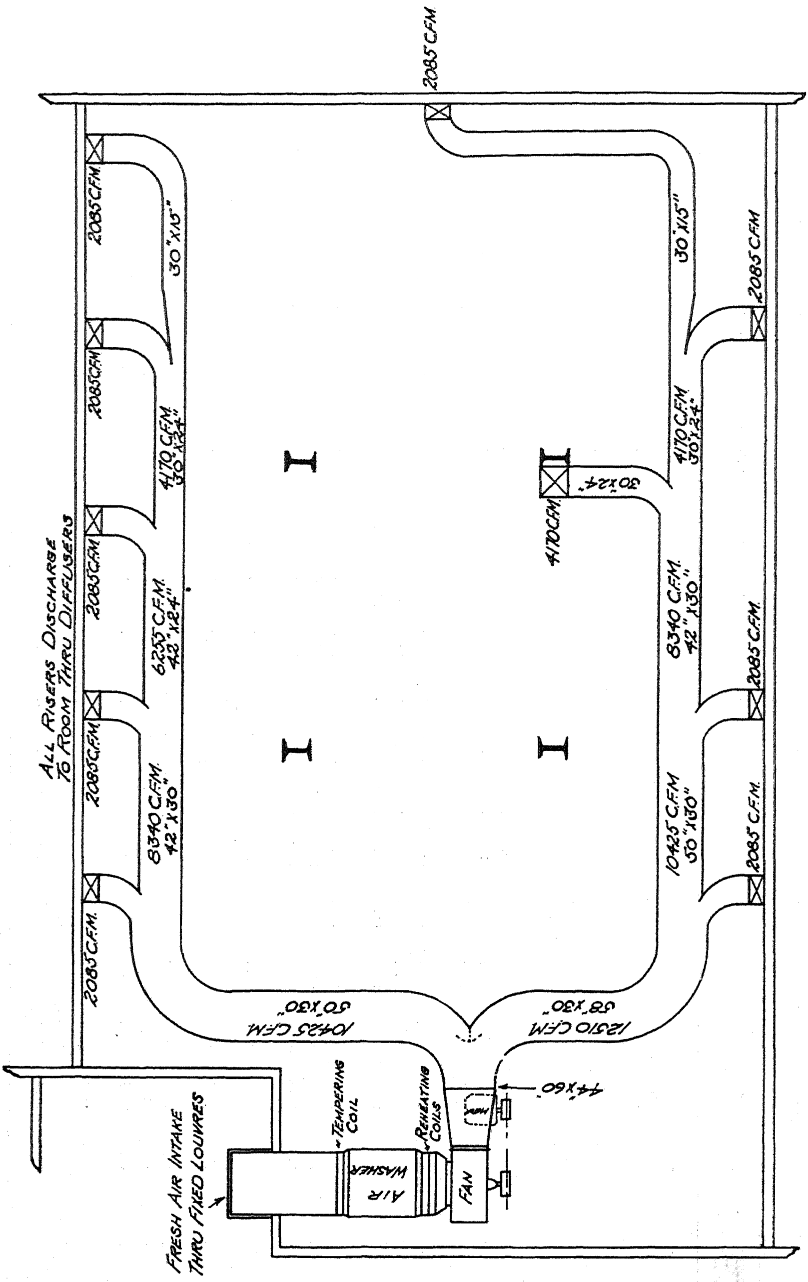

FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

22.94 sq ft. If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F. The washer with its automatic control will then raise the temperature from 32 F to 42 F. If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation. The reheating coil is selected to raise the temperature of the air from that leaving the air washer to 70 F. The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft. For more detailed information on tempering coil and air washer control, see Chapters 23 and 14.

Since the plan shows a moderately short run of main duct with no risers near the fan outlet, a fan should be selected which will have the required capacity of 22,935 cfm with a maximum velocity through the fan outlet of 1400 fpm. The outlet area, therefore, should be 16½ sq ft.

TABLE 2. PIPE SIZES FOR EXAMPLE 3[a]

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	100.0	56	60 x 44
12,510	54.6	45	58 x 30
10,425	45.4	42	50 x 30
8,340	36.3	39	42 x 30
6,255	27.2	35	42 x 24
4,170	18.2	29½	30 x 24
2,085	9.1	23	30 x 15

[a]Velocity through diffusers (not shown) to be approximately 300 fpm.

The main pipe size should be selected to give a velocity equal to or less than the velocity at the fan outlet. Choosing a 56-in. pipe with a cross-sectional area of 17.1 sq ft, the velocity in the main pipe will be 1340 fpm. Using the friction pressure loss method this 56-in. main pipe will be taken as the basis of calculation.

Fig. 6 shows the amount of air to be handled by each section of pipe. Expressing the volume handled by each section as a percentage of the total volume and using the charts, Figs. 4 and 5, the pipe sizes are as shown in Table 2.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 in. × 30 in. and 50 in. × 30 in. branches.

The rectangular equivalents are selected from Table 1; the width to depth proportion will be determined by construction requirements and ease of fabrication. The calculation of the friction is as follows:

The longest run from the fan outlet to diffuser is 150 ft 0 in.; 150 ft of 56-in. pipe is equivalent to $\frac{150 \times 12}{56}$ .... 32.2 dia.

Two 45-in., 90-deg elbows $(2 \times \frac{45}{56} \times 10)$ .... 16.1 dia.

Two 23-in., 90-deg elbows $(2 \times \frac{23}{56} \times 10)$ .... 8.2 dia.

Two 23-in., 90-deg elbows in riser $(2 \times \frac{23}{56} \times 30)$ .... 24.7 dia.

(Two bad elbows in riser, each equivalent to 30 diameters of duct).

Total diameter of 56-in. pipe 81.2

The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{81.2}{50} \times 0.112 = 0.182$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss. This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0.033 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in. The regain equals ⅔ (0.122 − 0.033) = 0.059 in.

The net static pressure loss in the duct only is then:

0.182 in. − 0.059 in. .... 0.123 in.

Other friction losses are as follows:

(1) Fresh air intake 1000-fpm velocity (1½ heads × 0.0625) .... 0.094 in.
(2) Tempering coil loss (from manufacturer's tables) .... 0.100 in.
(3) Air washer loss (from manufacturer's tables) .... 0.250 in.
(4) Reheating coil loss (from manufacturer's tables) .... 0.100 in.
(5) Allowance for regulating dampers and diffusers .... 0.100 in.

Static pressure loss of system .... 0.767 in.

The fan should be selected from the manufacturer's ratings which, according to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[1], will deliver 22,935 cfm at a static pressure of 0.767 in. and which has an outlet area of 16½ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at

TABLE 3. PIPE SIZES FOR EXAMPLE 4[a]

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21.6	16 x 24

[a]Velocity through intake grilles (not shown) to be approximately 400 fpm.

[1]See Chapters 17 and 41.

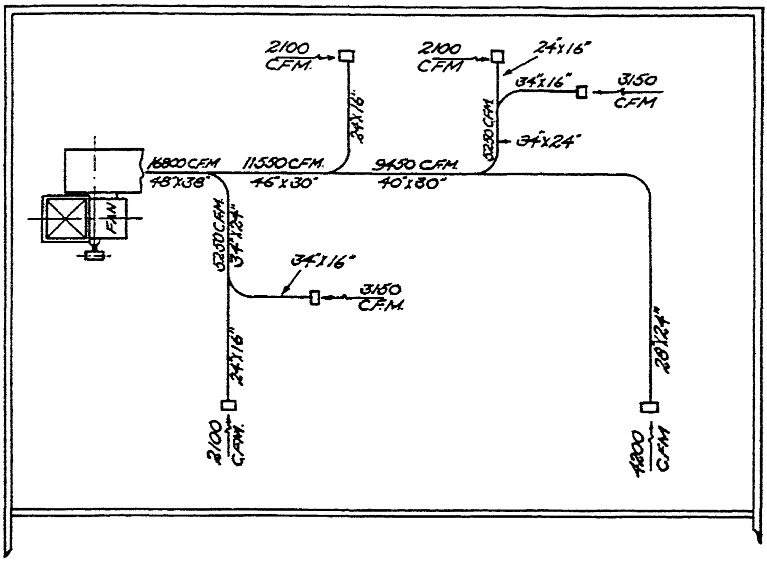

FIG. 7. EXHAUST SYSTEM LAYOUT

the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe. Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3.

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft.

(1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ .... 25.6 dia.

Two 28½-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ .... 36.4 dia.

(Two bad elbows in riser each equivalent to 30 diameters of duct).

One 28½-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 12}{47}\right)$ .... 6.0 dia.

Total diameter of 47-in. pipe.................... 68.0 dia.

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 122$ in.

Taking 50 diameters as one head loss, then $\frac{68 \times 0.122}{50}$ .... 0.166 in.

(2) Intake loss from grille (1½ heads at a 400 fpm velocity 1½ × 0.01) .... 0.015 in.
(3) Static pressure required to produce one velocity head at 1400 fpm.................... 0.122 in.
(4) Loss occasioned by step-up of velocity (0.20 × 0.122).................... 0.024 in.

(This loss varies from 0.05 to 0.40 velocity head depending upon the nature of the change. For average systems 0.20 velocity head is a close approximation.)

Static pressure loss on inlet side.................... 0.327 in.

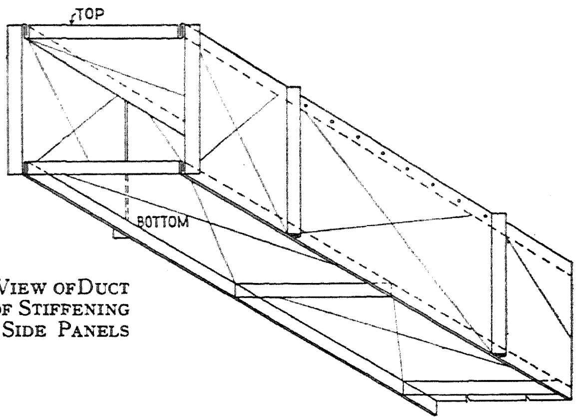

FIG. 8. ISOMETRIC VIEW OF DUCT SHOWING LOCATION OF STIFFENING SEAMS ON TOP AND SIDE PANELS OF DUCT

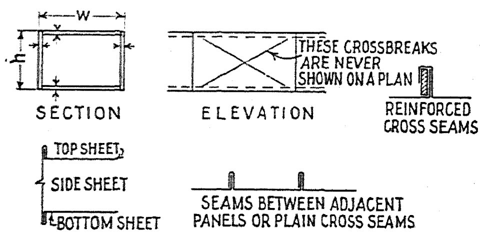

FIG. 9. DETAILS OF SEAMS

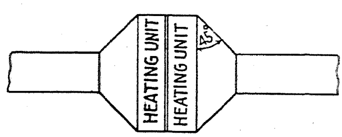

FIG. 10. METHOD OF INSTALLING HEATING UNIT

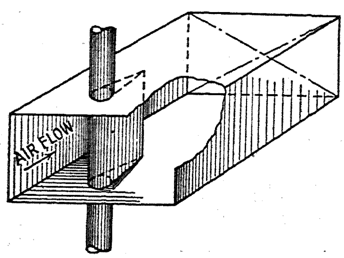

FIG. 11. INSTALLATION OF EASEMENT IN DUCT AROUND OBSTRUCTION

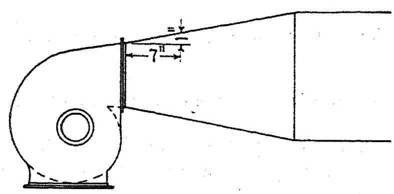

FIG. 12. FAN DISCHARGE CONNECTION

To this must be added the resistance on the discharge side of the fan. A fan outlet velocity of approximately 1500 to 1600 fpm may be used. Assuming the fan outlet to be equivalent in area to a 45-in. pipe, the velocity is 1525 fpm.

Loss on discharge (15 ft from fan outlet to discharge):

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in. pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0.145 and the discharge-side loss is $\frac{0.145 \times 4}{50} = 0.012$ in. The total static pressure loss of the system is then:

$$0.012 + 0.327 = 0.339 \text{ in.}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0.339 in. and to have an outlet velocity of 1525 fpm. Outlet area 11 sq ft.

Where there are one or more ducts with branches, the velocity of air in the ducts may be either chosen arbitrarily or calculated for friction losses. When arbitrary values are assigned, a certain amount of dampering should be provided for; this will be small when the method chosen permits a drop in velocity as the quantity of air is reduced.

After the total air quantity and the size of fan are ascertained, the main duct is usually fixed as being at least equal in area to the fan outlet, or perhaps 10 per cent greater. From this main pipe all others are proportioned. For example, if the main duct is 30 in. in diameter, a branch to carry 10 per cent of the total capacity should be 12.7 in. in diameter (see Fig. 4) in order to have the same friction per foot of length, while one carrying one-half the total capacity of a 30-in. main with the same friction loss per foot would be 23.4 in. in diameter. By this method of equalizing friction it is unnecessary to consider the resistance of each section of pipe independently, but only to know the distance from the fan outlet to the end of the longest run of pipe, the number and size of elbows, and the diameter and velocity in the largest pipe.

Example 5. If the greatest length of piping in a system is 130 ft with a 26-in. diameter main pipe and one 20-in. elbow, the piping having been designed for equal friction per foot of length, the friction would be the same as for 130 linear feet of 26-in. pipe, or 60 diameters. To this should be added the friction loss in elbows, in this case one 20-in. elbow, which has a loss equivalent to one-fifth of a velocity head or ten diameters of 20-in. pipe. This in turn is $\frac{20}{26} \times 10 = 7.7$ diameters of 26-in. pipe. The total equivalent length of the system will then be $60 + 7.7$, or 67.7 diameters. Since 50 diameters is equivalent to one velocity head, the loss is $\frac{67.7}{50} = 1.35$ times the velocity head. If the velocity is, for example, 2200 fpm, corresponding to 0.3-in. pressure, the friction loss of the system will be $1.35 \times 0.3 = 0.405$ in.

Frequently the prevention of sound in a heating or ventilating system imposes more severe restrictions than the prevention of excessive pressure drop. This question is highly involved and requires consideration of many factors. The air velocities to be used will vary with the standard of construction used in the ducts themselves as well as with the nature of the occupancy and the construction of the building. In general, architects and engineers who leave the details of duct construction to the contractor must, of necessity, design for lower velocities than might be required for quiet operation if proper construction details were always followed. The

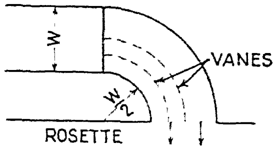

Fig. 13. Air Splitters Installed in Elbow

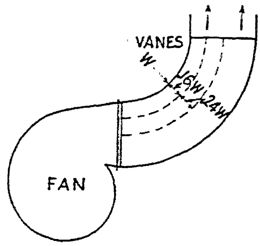

Fig. 14. Air Splitters Installed in Elbow at Fan Discharge

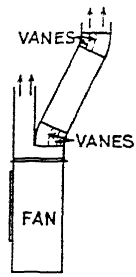

Fig. 15. Air Splitters in Branch Ducts and Elbows

contractor may be expected to build the ducts by the least expensive methods, and the engineer must anticipate this. For further information on noise reduction, see Chapter 18.

Details of Duct Construction

If panel construction is used with standing seams or similar reinforcement, and the panels are cross-broken to give rigidity, there is less likelihood of vibration due to air flow, or deflection due to air pressure. Elbows made without splitters, and improperly shaped transformation sections produce high local velocities which are the cause of noise in duct work. The use of first-class duct construction with well-designed transformation sections and splitters in elbows tends to maintain relatively uniform velocities with decrease in turbulence and in the noise produced.

Figs. 8 to 15 show acceptable construction details for rectangular ducts, elbows, transformation pieces or connections, and air splitters. Other methods are also acceptable, such as the use of angle iron stiffeners for large ducts. Good construction is essential to the elimination of duct noises and for the prevention of a flimsy installation.

Fig. 8 is an isometric view of a duct showing the location of the stiffening seams on the top and side panels. The cross seams should not occur at the same place but should be staggered as indicated. Heating units should be installed as shown in Fig. 10 with the duct connections making an angle of not less than 45 deg, but preferably 60 deg. Fan dis-

Table 4. Sheet Metal Gages for Rectangular Duct Construction[a]

Gage	Width of Duct	Seam	Reinforced Seam
26	Up to 12 in.		
24	13 in. to 30 in.	1	
22	31 in. to 48 in.	1	
22	49 in. to 60 in.	1½	⅛ in. x 1⅜ in.
20	61 in. to 90 in.	1½	⅛ in. x 1⅜ in.

[a]If panels are not cross-broken two gages heavier material should be used.

charge connections should have a maximum slope of 1 in 7, as indicated in Fig. 12. Whenever a pipe or other obstruction passes through a duct an easement should be placed around the pipe as indicated in Fig. 11. Air splitters should be installed in elbows as shown in Figs. 13 and 14. The recommended gages for rectangular sheet metal duct construction are given in Table 4.

REFERENCES

Fan Engineering, Buffalo Forge Co.

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The Flow of Liquids, by W. H. McAdams, *Refrigerating Engineering*, February, 1925, p. 279.

A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler, *A.S.M.E. Transactions*, Hydraulics Section, August, 31, 1933, p. 7.

PROBLEMS IN PRACTICE

1 ● Why is it desirable to make elbows with a radius equal to one and one-half times the pipe diameter?

Reference to Figs. 1 and 2 will show that while the loss of velocity head, as indicated by the curves, shows considerable variation for elbows between the range of 50 and 150 per cent radius, the line is practically straight after 150 per cent, indicating very little variation in loss of head for elbows of larger radius.

2 ● What is the best shape to use for ducts?

The shapes to be used in designing ducts, in the order of their preference, are round, square, and rectangular.

3 ● What determines which shape to use?

Structural and space conditions. Because ducts are as a rule part of the building or structure, it is necessary to proportion their sizes to fit the spaces available.

4 ● What is meant by "arbitrarily fix the velocity in the various sections?"

When using the velocity method as a basis for design, the maximum allowable velocity is fixed for the main supply duct at the fan, and this velocity is gradually decreased as each branch or outlet is taken off the main supply duct.

5 ● Which system of duct design is to be preferred, the velocity method or the friction pressure loss method?

The friction pressure loss method can be used to advantage where no structural or building conditions limit the shape of the ducts. Where these limiting conditions exist the velocity method is to be preferred.

6 ● Are the grille sizes figured on the same basis as the outlets?

The free area through the grilles is figured the same as the outlets, and this area is increased from 20 to 50 per cent, depending on the design of the grille, to allow for the loss of area caused by the construction of the face of the grille.

7 ● Where it is necessary to provide steel angle braces, how far apart should they be spaced?

Angle braces for large ducts should be placed on 3-ft 0-in. centers.

8 ● How much air will a 10-in. by 24-in. duct handle if it is part of a system designed on a pressure drop of 0.1 in. per 100 feet of run?

1450 cfm (Table 1 and Fig. 3).

9 ● How does a splitter at a duct junction influence the volume of the air going through each branch?

A splitter facing the direction of air flow cuts off the air and delivers the desired amount to the branch.

10 ● Why does a wide, shallow duct offer more resistance to the flow of air than does a square duct of equal cross-sectional area?

The perimeter of the wide, flat duct is greater than that of the square-section duct, so the former has the greater frictional area which increases the resistance and thus reduces the volume at any given pressure.

11 ● What methods are used to keep large ducts from vibrating because of air pulsations, and from sagging because of their own weight?

External bracing, such as standing seams, or structural shapes, like tees or angles, should be placed across the top and bottom. Exterior braces or cross buckling of metal sheets in diagonal panels may be used for the sides of large ducts.

12 ● What velocities of air flow should be used in the trunk ducts of a ventilating system in a public building?

From 1200 to 1600 fpm.

13 ● In a ventilating system in a residence, what is the recommended air velocity through supply registers and grilles?

400 fpm.

Chapter 21

INDUSTRIAL EXHAUST SYSTEMS

Types, Design of Systems, Suction and Velocity Requirements, Design of Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Selection of Fans and Motors

EXHAUST and collecting systems are found in almost every industry and are a vital adjunct in maintaining safe and hygienic conditions[1]. The present chapter attempts to give general information relating to the design of factory exhaust systems in order that efficient and economical control of dusts and fumes may be achieved.

TYPES OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at

[1]Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, July, 1934).

all, owing to the rising air current carrying the dust up through the breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN OF SYSTEMS

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[2]:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should not interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The design of an exhaust system should afford easy access to parts for inspection and care.

[2]For more detailed requirements see Safe Practice Pamphlets Nos. 32 and 37, published by the *National Safety Council*, Chicago.

SUCTION AND VELOCITY REQUIREMENTS

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the

TABLE 1. SIZE OF CONNECTIONS FOR WOOD-WORKING MACHINERY

TYPE OF MACHINE	DIAMETER OF CONNECTIONS IN INCHES
Circular saws, 12-in. diam.	4
Circular saws, 12-24-in. diam.	5
Circular saws, 24-40-in. diam.	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in.	5–6
With knives, 10-20 in.	6–8
With knives, 20-30 in.	6–10
Shapers, light work	4–5
Shapers, heavy work	8
Belt sander, belt less than 6 in. wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in.	5
Drum sander, 30 in.	6
Drum sander, 36 in.	7
Drum sander, 48 in.	8
Drum sander, over 48 in.	10
Disc sander, 24 in. diam.	5
Disc sander, 26-36 in. diam.	6
Disc sander, 36-48 in. diam.	7
Arm sander	4

direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

DIAMETER OF WHEELS	MAX. GRINDING SURFACE SQ IN.	MIN. DIAM. OF BRANCH PIPES IN INCHES
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in. thick	43	3½
10 in. to 16 in., " " " 2 in. "	101	4
17 in. to 19 in., " " " 3 in. "	180	4½
20 in. to 24 in., " " " 4 in. "	302	5
25 in. to 30 in., " " " 5 in. "	472	6
Buffing—		
6 in. or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in. thick	57	4
13 in. to 16 in., " " " 2 in. "	101	4½
17 in. to 20 in., " " " 3 in. "	189	5
21 in. to 27 in., " " " 4 in. "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

	STATIC SUCTION IN INCHES OF WATER
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2–4
Fur and felt machinery exhaust	2–3
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3–5

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but

it is safe to assume that for most dusty operations they should not be less than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al[3] give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

DESIGN OF HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis:

$$\frac{Y}{100 - Y} = \frac{0.1A}{x^2} \tag{1}$$

where

Y = per cent of velocity at opening.
A = area of opening, square inches (or square feet).
x = distance outward from opening, inches (or feet).

It is important to note that the velocity function varies in direct proportion to the area. Hence, under certain conditions, a large opening may function more effectively than a small one for the same volume of flow. The formula, of course, presumes that the air velocity distribution across the hood opening is uniform[4].

Example 1. A small hood 64 sq in. in area handles 400 cfm. What will be the air velocity at a point 5 in. outward along the axis if the flow is unobstructed?

[3]Hatch, Theodore, Drinker, Philip, and Choate, Sarah P., Control of the Silicosis Hazard in the Hard Rock Industries. I. A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

[4]Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS Vol. 38, 1932).

Solution. Substitute in Equation 1 and solve for Y, thus

$$\frac{Y}{100 - Y} = \frac{0.1 \times 64}{5 \times 5}$$

from which $Y = 20.4$ per cent of the velocity at the opening of the hood.

$$\text{Velocity at opening} = \frac{400 \times 144}{64} = 900 \text{ fpm}$$

Hence, the velocity at the point in question is $900 \times 0.204 = 184$ fpm

Air Flow from Static Readings

The volume of air flow into any hood may be determined from the following equation:

$$Q = 4005\, fa \sqrt{h_t} \qquad (2)$$

where

Q = volume of air flow, cubic feet per minute.
a = area of connecting duct, square feet.
h_t = static suction at throat of hood, inches of water.
f = orifice or restriction coefficient, which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. If it is assumed that the entrance loss of a hood is proportional to the velocity head, f can be determined by the relation:

$$f = \sqrt{\frac{h_v}{h_v + h_e}} \qquad (3)$$

where

h_v = the velocity head.
h_e = the entrance loss.

For duct ends and abrupt openings $h_e = h_v$ and for flared openings $h_e = 0.5h_v$.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity function at any point along the axis varies directly as the area of the opening and inversely as the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be subdivided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12-in. elevation. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of

the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \qquad (4)$$

where

Q = total volume of air handled by hood, cfm.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, fpm.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept below 100 parts per million. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located in a horizontal position slightly above the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with a vertical exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth and the vapors should be discharged through a suitable stack to permit dilution[5].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is opened to its maximum height.

DESIGN OF DUCT SYSTEMS

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and par-

[5]For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926, Harrisburg, Pa.

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

At the point of entrance of a branch pipe with the main duct, there

should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust	2000
Wood chips and shavings	3000
Sawdust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5a and 5b may be used as tests to determine the conveying efficiency of a system[6]. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:

$$V = 13{,}300 \frac{s}{s+1} d^{0.570} \tag{5a}$$

For horizontal ducts:

$$V = 6000 \frac{s}{s+1} d^{0.398} \tag{5b}$$

where

V = air velocity in duct, feet per minute.
s = specific gravity of particles.
d = average diameter of largest particles conveyed, inches.

Example 2. Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog (0.57 × log 0.37) = 0.568; the required velocity is, therefore, 5500 fpm. Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

[6]DallaValle, J. M.: Determining Minimum Air Velocities for Exhaust Systems. (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, September, 1932).

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 3, Chapter 20. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \tag{6}$$

where

h_c = the pressure drop through the cyclone, inches of water.
V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts, feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filter cloths are used when the material collected by an exhaust system is valuable or cannot be separated from the air with an ordinary cyclone.

They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are slightly lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For particular features concerning special fans, consult the *Catalog Data Section* of THE GUIDE and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated.

Further information regarding the choice of fans and motors is given in Chapter 17.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER	LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200 to 300	14

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 7. Hoods and ducts when short, may frequently be constructed of wood and be quite effective.

TABLE 7. MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION[a]

TYPE OF FUME CONVEYED	PROTECTIVE MATERIAL TO BE USED
Chlorine	Rubber lining or chrome-nickel alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickel alloys
Hydrochloric acid	Rubber lining, chrome-nickel alloys
Nitrous gases	Nickel-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol. 24, 1932).

Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

PROBLEMS IN PRACTICE

1 ● Should individual operations be served by an individualized dust collector system?

Yes, if operations are usually kept individual in a group of machines.

2 ● Are state regulatory requirements as to *suction* applicable to all sorts of dust collecting installations?

As a rule the regulations refer only to grinding wheel and buffing wheel systems. They are needed for many other industrial processes.

3 ● What is the most common method of reducing total air volumes handled in cases employing large hoods over apparatus covering a large area?

The use of the petticoat or double hood which permits a comparatively high air velocity at the rim of the hood and controllably small velocities in the center.

4 ● What other types of collectors are available for use in the place of cyclones and filters when chemical and physical conditions obviate the possibility of the use of them?

Devices such as scrubbers and contactors, using water or other contacting liquids, electrical precipitators, and dynamical precipitators.

5 ● What is the most frequent error made in dust collector system design?

The omission of some means of putting into the workroom air having the proper characteristics to replace that which has been exhausted.

6 ● Are there available means for testing the performance of dust collecting systems when they are required to meet high industrial hygienic standards?

Yes. Such means are set up by the United States Public Health Service and by the Standard Code for Testing Centrifugal Fans (Chapter 41).

7 ● Why is it not permissible to connect up emery wheels and buffing wheels to the same exhaust system?

Emery wheels and buffing wheels should be handled by separate systems because of the fire hazard, as it is possible for sparks from the emery wheels to ignite the lint and dust from the buffing wheels when both are carried through the same system.

8 ● Give an important characteristic of centrifugal type dust collectors which should be given consideration when applying this type of collector to installations requiring high separating efficiencies.

The separating action of a cyclone or centrifugal type collector depends largely on centrifugal force. Reducing the radius of air flow increases the centrifugal force for a given velocity of flow. Accordingly, the smaller size units usually give higher separating factors, and better results can sometimes be obtained by using a number of small collectors instead of one large unit.

9 ● Mention some general suggestions relating to the design of efficient industrial exhaust systems.

a. Endeavor to obtain a maximum degree of effectiveness with a minimum volume of air, by the use of well designed hoods closing in the sources of fumes or material to be removed so located as to take advantage of the natural direction taken by the fumes or materials when leaving their source.

b. Give particular care to the velocity of flow. The duct velocities for material conveying systems must be high enough to properly carry the material, but they should not be higher than necessary because excessive velocities increase the pressure requirements and result in a waste of power.

c. Select the type of fan best suited to the job. For installations where stringy material is handled do not use a fan wheel which has a shroud.

d. When handling the refuse from various machines, study the grouping and operating cycles of the machines. Connecting a large number of machines into one system is frequently very uneconomical.

e. Avoid unnecessary distances and bends in laying out the piping system.

10 ● The static pressure measured at the throat of a buffing wheel hood is 2 in. and the velocity head measured with a Pitot tube is 1.6 in. Calculate the restriction coefficient f.

From Equation 2, $V = 4005 f \sqrt{h_t}$.

From the theory of air flow, $V = 4005 \sqrt{h_v}$.

Hence, $\sqrt{h_v} = f \sqrt{h_t}$.

or

$$f = \sqrt{\frac{h_v}{h_t}} = \sqrt{\frac{1.6}{2.0}} = 0.89$$

11 ● A tank, 4 ft by 8 ft, contains a fluid which gives off injurious vapors. A large hood is located 30 in. above the top of the tank and extends slightly over its edges. Assuming that a velocity of 60 fpm is required to adequately control the vapors near the edges of the tank, calculate the air flow required.

Using Equation 4, $P = 2 \times 4 + 2 \times 8 = 24$ ft; $D = 30$ inches $= 2.5$ ft; $V = 60$ fpm.

$$\text{Hence, } Q = 1.4 \times 24 \times 2.5 \times 60 = 5040 \text{ cfm.}$$

12 ● Silica dust with a specific gravity of 2.65 is being conveyed in a duct system. The velocity measured in a vertical portion of the system is found to be 2700 fpm. What is the maximum diameter particle transported at this velocity?

$$\text{Using Equation 5a, } 2700 = 13{,}300 \times \frac{2.65}{3.65} \times d^{0.570}$$

from which

$$d = (0.28)^{1.75} = 0.11 \text{ in.}$$

Chapter 22

FAN SYSTEMS OF HEATING

Types of Systems, Blow-Through, Draw-Through, Heating Units, Design, Temperatures, Weight of Air to be Circulated, Temperature Loss in Ducts, Heat Supplied Heating Units and Washer, Grate Area, Boiler Selection, Weight of Condensate, Static Pressure, Fans and Control

A FAN system of heating depends upon fans and blowers to distribute air through ducts from one centrally located plant. This chapter considers heating and humidifying systems of this type whereas similar systems arranged for cooling and dehumidifying are discussed in Chapter 9. A special type of central fan system, the mechanical warm air or fan furnace system, which is especially adapted to residences, churches, halls, and other small buildings, is covered in Chapter 23.

TYPES OF SYSTEMS

In the indirect type of central fan heating and air conditioning systems, steam is usually the medium by which heat is transferred from the boiler, or other source of heat, to the heating units. If the system is intended solely for heating, the air is passed over one or more stacks or batteries of heating units and then conveyed to the spaces for which it is intended through a system of ducts. In some cases, a predetermined amount of outside air is introduced for ventilating purposes, whereas in others the moisture content is controlled by passing the air through a washer or humidifier. If the apparatus is designed to control simultaneously the temperature, humidity, air motion, and distribution, it is known as an air conditioning system.

In the *split system*, the heating is accomplished by means of radiators or convectors, and the ventilating or air conditioning by means of the central fan apparatus. In the *combined system*, the entire operation of heating, ventilating, and air conditioning is handled by the central fan system.

A common arrangement of the central fan system of heating is illustrated by Fig. 1 and consists of a fan, a heating unit (heater) enclosed by a sheet metal casing connected with the suction side of the fan, a sheet metal casing connected to the heating unit casing run to the outside of the building and provided with an adjustable opening inside the building for recirculation of the air when desired, and a duct system attached to the fan outlet to convey and distribute the air to various parts of the building to be warmed by the apparatus. The fan is ordinarily motor-driven; there are, however, many cases when a direct-connected steam engine may be used to advantage. In this event the exhaust from the engine can be con-

nected to one or more sections of the heater, depending upon the condensation rate of the engine. The recirculation duct connected with the opening in the suction duct should be extended to a point as near the floor as possible.

When ventilation is not a requirement or is considered relatively unimportant, as in shop and factory heating, and the number of persons vitiating the air is small compared with the cubical contents of the building, or the process does not generate obnoxious gas or vapors, the air may be recirculated, sufficient outside air for ventilation being supplied by infiltra-

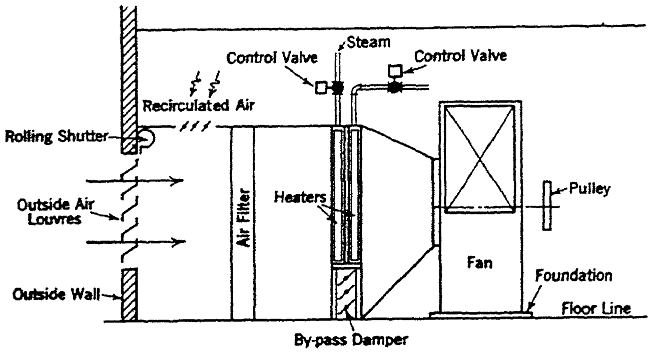

Fig. 1. Arrangement of a Central Fan Heating System (Draw-Through)

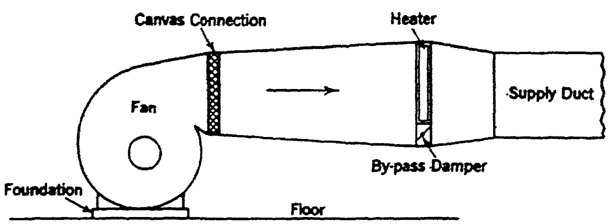

Fig. 2. Arrangement for Heating Unit (Blow-Through)

tion. The amount of heat to be supplied the heating unit in this case is the same as would be required for a direct radiation installation.

When ventilation is a requirement to be met, an arrangement similar to that shown by Fig. 1 may be employed. Since the amount of air necessary for heating is generally in excess of the amount required for ventilation, considerable fuel economy may be effected by recirculating a portion of the air. In this case only sufficient outside air is drawn into the system to meet the ventilation requirement and the remainder of the air, required for heating, is recirculated. This may be readily effected by an arrangement of ducts and dampers on the suction side of the fan as previously mentioned. If the outside air introduced is to be washed or conditioned the washer or humidifier and tempering coil may be added between the inlet for the recirculated air and the fresh air intake.

Blow-Through, Draw-Through

When the heating unit is located on the suction side of the fan, the system is known as *draw-through*. (See Fig. 1.) When the heating unit is located in the discharge from the fan, the system is known as *blow-through*. (See Fig. 2.) The draw-through combination is used for factory and toilet room installations because a more compact arrangement of the apparatus usually is possible. In addition, air leakage will be inward. The blow-through combination is used principally in schools and public buildings, and for all booster coil arrangements where different temperatures and independent temperature regulation are required for different heated spaces. In public building installations, the fan frequently blows the heated air into a plenum chamber from which the air ducts radiate to the various rooms of the building; this arrangement is sometimes called the *plenum* system.

HEATING UNITS

The heating units for central fan systems using steam as the heating medium may be classified as (1) tempering coils, (2) preheater coils, (3) reheater coils, (4) booster coils, and (5) water heaters, either open or closed. *Tempering coils* are used with ventilating and air conditioning systems for raising the temperature of the outside cold air to above freezing, or 32 F. They are not required for heating systems where all of the air is recirculated, since the temperature of the recirculated air will be above freezing. *Preheater coils* are used with air conditioning systems to raise the temperature of the air from that leaving the tempering coils to such a temperature that in passing through the water sprays of the washer (without water heater) the air will become partially saturated (adiabatically) having a moisture content corresponding to the required dew-point temperature. Preheater coils therefore supply heat as necessary to control the dew-point temperature. The *reheater coils* are used to raise the temperature of the air leaving the tempering coils (in the case of a heating or ventilating system) or the air leaving the washer (in the case of an air conditioning system) to that necessary to maintain the desired temperature in the rooms or spaces to be heated or conditioned, except where booster coils are used, in which case the reheater coils raise the air temperature to approximately room temperature, or slightly higher. *Booster coils* are installed in the duct branches to control the temperature of the air entering the rooms or spaces for which it is intended. *Water heaters* are used on an air conditioning system to control the dew-point temperature. They are used mainly for industrial work, seldom for comfort conditioning. They are not used where preheater coils are employed. The open type supplies steam directly to the spray water, while the closed type utilizes a heat interchanger by which the steam imparts its heat to the spray water. Where water heaters are required for comfort conditioning, the closed type is used.

The heating units for central fan systems in use at the present time consist either of pipe coils, finned tubes of steel, copper, brass or other metal, cast-iron sections with extended surfaces, or the cellular type. Steam is passed through these heating units and the air to be heated is passed over their exterior surfaces.

In selecting a heating unit for any particular service, the choice should be based on the desired requirements as follows:

1. Final temperature desired.
2. Loss in pressure (or friction) of air passing over the heating unit.
3. Air velocity over the heating unit.
4. Free area or face area of heating unit.
5. Ratio of heating surface to net free (or face) area.
6. Air volume required.
7. Number of rows of pipes, tubes, or sections.
8. Amount of heating surface.
9. Steam pressure drop through the heating unit.
10. Weight of heating unit.

Final Temperature Desired. The choice of a heating unit is largely influenced by the final temperature desired, when the entering air temperature and steam pressure available at the heating unit are specified. These data are obtainable from manufacturers' catalogs.

Loss in Air Pressure (or Friction). The allowable friction through the heating unit is one of the first factors to be determined in the selection of the apparatus. The velocities of air through various types of heating units will not necessarily be the same, but for any particular job the velocity through the heating unit should be a secondary consideration and the allowable friction or air pressure loss should be fixed approximately before proceeding with the selection of the heating unit. The loss in air pressure (or friction) through the heating unit should not exceed a predetermined maximum allowable amount for economical operation and for moderate size and first cost of installation.

In public building work, the maximum allowable friction through both tempering coil and reheater coils should never exceed ⅝ in. of water and it is advisable that the friction be kept considerably lower than this figure if possible. A tempering coil friction ranging from 0.10 to 0.20 in. of water is considered satisfactory. The air pressure loss for reheaters ordinarily ranges from 0.20 to 0.40 in. of water. In factory work, the maximum friction through the heater should never exceed 0.8 in. or 1 in. of water and it is advisable to figure the heaters at lower frictions if possible.

Velocity through Heating Unit. This velocity has generally been given in manufacturers' tables as being measured at 70 F and in most cases refers to the velocity through the net free area of the heating unit, or through the net space between the pipes, tubes or sections. Although most manufacturers give suitable velocities measured at 70 F, certain manufacturers show velocities measured at 65 F and others indicate velocities measured at the average air temperature through the heating unit. Many new heating units, however, specify net face areas with corresponding velocities instead of velocities through net free areas. In either case, manufacturers publish the corresponding friction or air-pressure loss in tables. The velocity through the net free area of the heating unit averages about 1000 fpm and that through the net face area about 500 fpm.

The *volume of air* to be heated in any particular case is determined after consideration of the ventilation requirements, heat losses, and quantity of air required for proper circulation, as explained in Chapters 2 and 7.

The number of *rows of pipes, tubes, or sections* or the amount of *heating surface* to be used may be selected from manufacturers' catalogs after the quantity of air handled and the heat load are known. Savings in operating expense or cost of installation should result from a proper selection of heater and by-pass areas. For example, instead of having the entire air quantity go through a one-row heating unit, it may be advantageous to use a two-row heating unit and a properly sized by-pass. Thus, when no heating is being done, a suitable by-pass damper may be opened to place a lighter load on the fan.

The *steam pressure drop through the heating unit* is also tabulated in manufacturers' data tables. The sizing of steam supply and return piping, allowing for drops through heating units, is explained in Chapter 32.

Weight of Heating Unit. In the design of a heating system, the weight limitations of heating units are determined by the location of the units. Obviously, if there is no loading limitation imposed, any type of heating unit may be selected. On the other hand if the heating unit is to be hung from the ceiling, it may be desirable to use the lightest unit which will accomplish the work required.

DESIGNING THE SYSTEM

The general procedure for the design of central fan systems is as follows:

1. Calculate the heat loss for each room or space to be heated.
2. Determine volume of outside air to be introduced.
3. Assume or calculate temperature of air leaving registers or supply outlets.
4. Calculate weight of air to be circulated.
5. Estimate temperature loss in duct system.
6. Calculate heat to be supplied the heating units and washer.
7. Select heating units and washer from manufacturers' data and performance curves.
8. Calculate total heat to be supplied.
9. Calculate grate area and select boiler.
10. Design duct system.
11. Calculate total static pressure of system.
12. Select fan, motor, and drive.

The heat losses (H) should be calculated in accordance with the procedure outlined in Chapter 7. If a positive pressure is maintained by the central fan system in the room or space to be ventilated or conditioned, there will ordinarily be very little infiltration of cold outside air through the cracks and crevices of the space. Consequently, the volume of air introduced into the space at the assumed or calculated outlet temperature need only be sufficient to provide for the transmission losses, plus about one-third of the infiltration losses. The exfiltration of heated or conditioned air through the cracks and crevices of the space should be provided for by making the usual allowance for the infiltration losses in arriving at the total heat loss of the space. The air required to make up for this exfiltration of heated or conditioned air will be brought in at the outside air intake and may be included as a part of the outside air neces-

sary for the ventilating requirements. The heat required to raise this air to the conditions maintained in the room must be provided by the tempering coils, preheater coils, and reheater coils. If a positive pressure is not maintained in the room or space to be conditioned, the normal infiltration of outside cold air will take place in this room, and the outlet temperature, together with the required air volume at this temperature, must be sufficient to provide for both infiltration and transmission losses.

Volume of Outside Air

The volume of outside air required for ventilation or air conditioning purposes may be determined from data in Chapter 2. In no case shall less than 10 cfm per person be introduced.

The heat required to warm the outside air introduced for ventilation purposes (H_o) may be determined by means of the following formula:

$$H_o = 0.24\ (t - t_o)\ M_o \tag{1}$$

where

0.24 = specific heat of air at constant pressure.
t = room temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
M_o = weight of outside air to be introduced per hour, in pounds = d_oQ_o.
Q_o = volume of outside air to be introduced, cubic feet per hour.
d_o = density of air at t_o, pounds per cubic foot.

Example 1. A building in which the temperature to be maintained at 70 F requires 10,000 cfm. If the outside temperature is 20 F, how much heat will be required to warm the air introduced for ventilation purposes to the room temperature?

Solution. $Q_o = 10{,}000 \times 60 = 600{,}000$ cfh; $d_o = 0.08276$ (Table 3, Chapter 1); $M_o = 0.08276 \times 600{,}000 = 49{,}656$ lb; $t = 70$ F; $t_o = 20$ F; $H_o = 0.24 \times (70 - 20) \times 49{,}656 = 595{,}872$ Btu per hour.

Temperature of Air Leaving Registers

If the system is to function only as a heating system, that is, entirely as a recirculating one, the temperature of the air leaving the register outlets must be assumed. For public buildings, these temperatures may range from 100 to 120 F, whereas for factories and industrial buildings the outlet or register temperature may be as high as 140 F. In no case should the outlet temperature exceed these values.

For ventilating or conditioning systems, the temperature of the air leaving the supply outlets may be estimated by means of the following formula:

$$t_y = \frac{55.2H}{Q} + t \tag{2}$$

where

t_y = outlet temperature, degrees Fahrenheit.
H = heat loss of room or space to be conditioned, Btu per hour.
Q = *total* volume of air to be introduced at the temperature t, cubic feet per hour.

If the outlet temperature (t_y) as determined from Equation 2 exceeds 120 F for public buildings, or 140 F for factories or industrial buildings,

these respective outlet temperatures should be used as factors in the following equation to determine the volume of air to be introduced into the room or space:

$$Q = \frac{55.2H}{(t_y - t)} \tag{3}$$

Example 2. The heat loss of a certain auditorium to be conditioned is 100,000 Btu per hour. The ventilating requirements are 90,000 cu ft per hour and the room temperature 70 F. Determine the outlet temperature.

Solution. Substituting in Formula 2,

$$t_y = \frac{55.2 \times 100{,}000}{90{,}000} + 70 = 131.3 \text{ F}$$

Inasmuch as this temperature is excessive, it will be necessary to assume an outlet temperature, which will be taken as 120 F, and to calculate the amount of air to be introduced into the room at this temperature to provide for the heat loss. Substituting in Equation 3,

$$Q = \frac{55.2 \times 100{,}000}{120 - 70} = 110{,}400 \text{ cfh (at temperature } t)$$

Weight of Air to be Circulated

The total weight of air to be introduced into the room or space to be heated or conditioned (M) is given by the following formulae:

$$M = \frac{H}{0.24(t_y - t)} = dQ \tag{4}$$

$$M = M_o + M_r \tag{5}$$

$$M_o = d_o Q_o \tag{6}$$

where

d = density of air at temperature t, pounds per cubic foot.
d_o = density of air at temperature t_o, pounds per cubic foot.
Q_o = volume of outside air at temperature t_o.
M_o = weight of outside air, pounds.
M_r = weight of recirculated air, pounds.

Example 3. Using the data of Example 2 and an outside temperature of 20 F, what will be the values of M, M_o and M_r?

Solution. $d = 0.07495$; $d_o = 0.08276$; $Q = 110{,}400$; $Q_o = 90{,}000$; $H = 100{,}000$.

$$M = \frac{100{,}000}{0.24 \times (120 - 70)} = 8{,}333 \text{ lb}$$

$$M_o = 0.08276 \times 90{,}000 = 7{,}448 \text{ lb}$$

$$M_r = M - M_o = 8{,}333 - 7{,}448 = 885 \text{ lb}$$

Temperature Loss in Ducts

The allowances to be made for loss in transit through the duct system (t_z) are as follows:

1. When the duct system is located in the enclosure to which the air is being delivered, as in a factory, it may be assumed that there is no loss between the reheater coil and the point or points of discharge into the enclosure.

2. For ducts in outside walls or attics, or other exposed places, allow 0.25 F per linear foot of uninsulated duct.

3. For ducts run underground an allowance shall be made based on the estimated heat loss of the duct, assuming the average temperature of the ground to be 55 F.

Heat Supplied Heating Units and Washer

The following cases may arise in practice:

A. The heating of the building is done entirely by means of a central fan system, all of the air being drawn from the outside.

B. Similar to *A*, except that all of the air is recirculated.

C. A portion of the air is recirculated, and the remainder is drawn in from the outside.

D. Air at the same temperature is to be delivered to all the rooms. A constant relative humidity is maintained in the building and all of the air circulated is drawn from outside the building. (Not applicable to the heating of various rooms where individual control of each room is desired.)

E. Outside air, return air, and by-pass air are used with the reheater located in by-pass air chamber.

F. Arrangement of apparatus where individual control of the temperature for each room is required in conjunction with air washer equipment to maintain a constant relative humidity in the rooms. The air washer is provided with a water heater for the spray water, capable of fully saturating the air. A section of preheater may be used for this purpose in place of the water heater. With this arrangement and with a uniform temperature of air entering the rooms, it is impossible to maintain the same room temperature throughout the building because the weight of air to be delivered to each room is determined and fixed by the ventilating requirements.

In analyzing these cases, the following symbols will be used:

H = heat loss of the room or building, Btu per hour.

H_1 = heat to be supplied to the reheater coil, Btu per hour.

H_2 = heat supplied tempering coil, or tempering coil and preheater, Btu per hour.

H_3 = heat supplied air washer by water heater, Btu per hour.

H_4 = heat to be supplied booster coil, Btu per hour.

M = weight of air to be introduced into the room or building, pounds per hour.

M_r = weight of recirculated air, pounds per hour.

M_b = weight of air by-passing washer, pounds per hour.

M_o = weight of air drawn in from outside, pounds per hour.

t_o = mean temperature of outside air, degrees Fahrenheit.

t = mean air temperature to be maintained in the room or building, degrees Fahrenheit.

t_1 = mean temperature of the air entering the reheater coil.

t_2 = mean temperature of the air leaving the reheater coil.

t_z = temperature loss in the duct system.

t_y = temperature of the air leaving the duct outlets.

t_x = average temperature of air entering tempering coil.

t_w = temperature of air entering washer.

0.24 = specific heat of air at constant pressure.

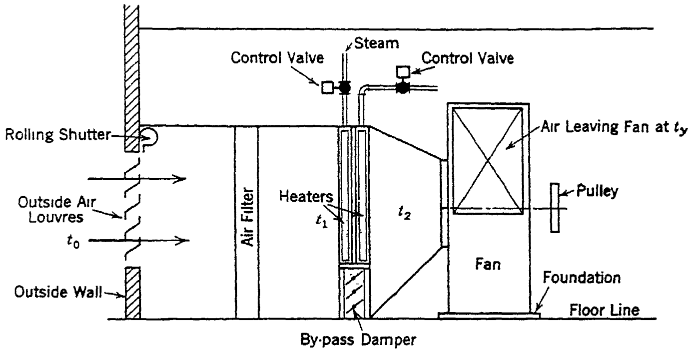

FIG. 3. HEATING UNIT AND FAN ARRANGED FOR OUTSIDE AIR CIRCULATION (*Case A*)

Case A. (Fig. 3) All of the air circulated to be drawn from outside the building, in which case $t_x = t_0$.

$$H_2 = 0.24\ (t_1 - t_0)\ M_0 \qquad (7)$$

$$H_1 = 0.24\ (t_2 - t_1)\ M_0 \qquad (8)$$

Example 4. The heat loss H for a certain factory building is 700,000 Btu per hour. The mean inside temperature t to be maintained is 65 F. The assumed outside air temperature t_0 is 0 F; $t_z = 0$, $t_y = t_2$ and is assumed to be 140 F. The temperature leaving the tempering coil is assumed to be 35 F. Required, H_1 and H_2. From Equation 4,

$$M = \frac{700{,}000}{0.24\ (140 - 65)} = 38{,}889 \text{ lb per hour.}$$

$H_2 = 0.24 \times (35 - 0) \times 38{,}889 = 326{,}667$ Btu per hour.
$H_1 = 0.24 \times (140 - 35) \times 38{,}889 = 980{,}003$ Btu per hour.
$H_2 + H_1 = 326{,}667 + 980{,}003 = 1{,}306{,}670$ Btu per hour.

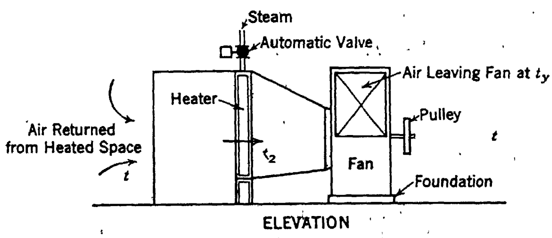

FIG. 4. ARRANGEMENT FOR RECIRCULATION (*Case B*)

Case B. (Fig. 4) All of the air is to be recirculated, in which case $t_1 = t$.

$M_r = 38{,}889$ lb
$M_1 = 0.24\ (t_2 - t_1)\ M_r$
$H_1 = 0.24\ (140 - 65) \times 38{,}889 = 700{,}000$ Btu per hour.

This example illustrates the saving in fuel consumption by the recirculation of the air. The heat to be supplied the apparatus is the same as that required for a direct system of heating and is equal to the heat loss of the building ($H_1 = H$), in the example 700,000 Btu per hour as compared with 1,306,670 for *Case A*.

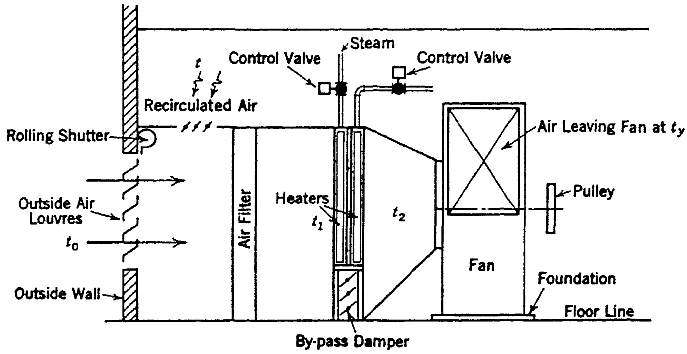

FIG. 5. COMBINATION OF RECIRCULATED AIR AND OUTSIDE AIR (*Case C*)

Case C. (Fig. 5) A portion of the air circulated is recirculated air and the remainder, as may be required for ventilating purposes, is drawn in from the outside. According to Equations 4 and 5,

$$M = M_o + M_r = \frac{H}{0.24\ (t_y - t)}$$

The temperature of the resulting mixture of outside and recirculated air entering the tempering coil is:

$$t_x = \frac{M_o t_o + M_r t}{M} \qquad (9)$$

Example 5. Assuming that a positive supply of outside air (d_o = 0.0864) is required for ventilation at the rate of 90,000 cu ft per hour in the preceding example, then M_o = 0.0864 × 90,000 = 7776 lb per hour are required, measured at 65 F.

$$M_r = M - M_o = 38{,}889 - 7776 = 31{,}113 \text{ lb}$$

$$t_x = \frac{7776 \times 0 + 31{,}113 \times 65}{38{,}889} = 52 \text{ F}$$

$$H_1 = 38{,}889 \times 0.24\ (140 - 52) = 821{,}336 \text{ Btu.}$$

This amount of work may be accomplished with one or more banks of heating units, that is, either a single reheater or a tempering coil and reheater.

The three preceding cases refer to installations in which conditioning the air to maintain certain relative humidity requirements does not enter into the problem, as for example, certain types of industrial installations. In practically all modern public buildings, theaters, schools, and in many industrial installations, the ventilating requirements include the provision for washing and humidifying the air delivered to the various rooms of the structure.

In the following cases it is assumed that in addition to maintaining a mean room temperature t, the heating and ventilating apparatus is required to maintain a constant relative humidity in the rooms.

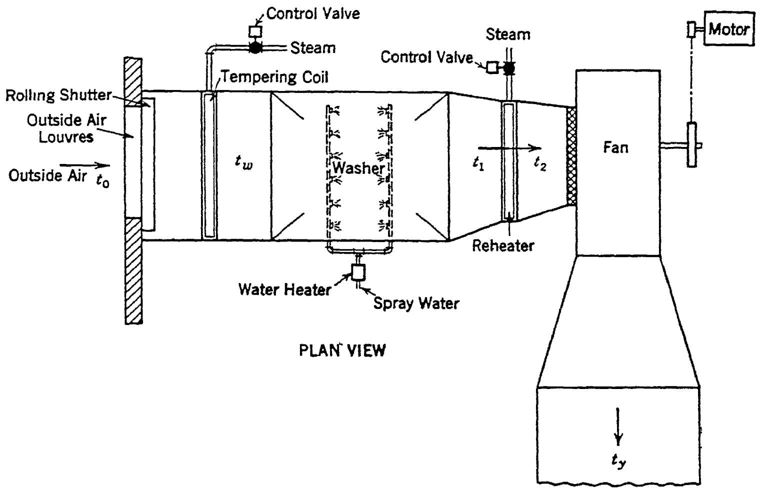

FIG. 6. OUTSIDE AIR CIRCULATED; CONSTANT RELATIVE HUMIDITY IN ROOM (*Case D*)

Case D. (Fig. 6) The maximum relative humidity that may be maintained within the building without the precipitation of moisture on single glazed sash when the outside temperature is 30 F is approximately 35 per cent. If the inside temperature t is 70 F, 35 per cent relative humidity corresponds to a dew-point temperature of 41 F. (See psychrometric chart.)

The installation shown in Fig. 6 contemplates the use of a tempering coil, an air washer provided with a water heater, and a reheater. The tempering coil, one section in depth, warms the incoming air to approximately 35 F to prevent freezing any of the spray water. The air passing through the spray chamber is saturated and leaves at a temperature of $t_1 = 41$ F.

The heat to be supplied the reheater is:

$$H_1 = 0.24\ (t_2 - 41)M \text{ Btu per hour.}$$

The heat to be supplied the tempering coil is:

$$H_2 = 0.24\ (35 - t_0)M \text{ Btu per hour.}$$

The amount of heat, per pound of air circulated, to be supplied the humidifying washer or humidifier is the difference between the heat content of the assumed dry air entering the washer at a temperature of $t_w = 35$ F and the leaving saturated air at $t_1 = 41$ F (Chapter 1), or:

$$15.7 - 8.4 = 7.3 \text{ Btu per pound of dry air.}$$

The amount of heat required for the washer is:

$$H_3 = 7.3\ M \text{ Btu per hour.}$$

The total amount of heat required by the apparatus is, therefore:

$$H_1 + H_2 + H_3 \text{ Btu per hour.}$$

If a washer having a *humidifying efficiency* of 67 per cent *without water heater* is employed it will be necessary to heat the outside air drawn into the apparatus by means of the tempering and preheater coils to such a temperature that the air in passing through

the water sprays will become partially saturated (adiabatically) having a moisture content per pound of air equal to saturated air at 41 F. If the incoming air is warmed to t_w = 88 F (requiring a two-section-depth heating unit) it will be cooled in the washer to 64 F, with a temperature drop of 88 − 64 = 24 deg.

If the *humidifying efficiency* of the washer were 100 per cent, the air would become adiabatically saturated at 52 F after a temperature drop of 88 − 52 = 36 F. The efficiency of the washer is, however, only 67 per cent, so that the actual temperature drop will be 0.67 × 36 deg or 24 deg, as used.

The heat to be supplied the reheater is in this case $H_1 = 0.24\ (t_2 - 64)\ M$ Btu per hour, and the heat to be supplied to the tempering coil and preheater is $H_2 = 0.24\ (88 - t_o)\ M$. The total heat required by the apparatus is $H_1 + H_2$, no heat being supplied to the washer.

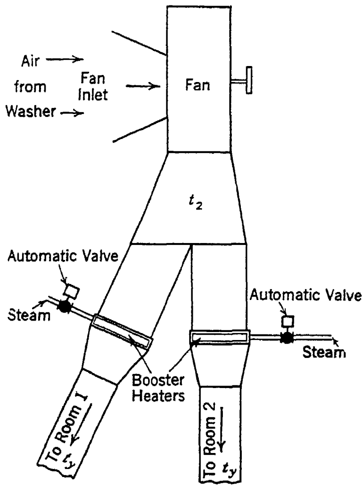

FIG. 7. OUTSIDE AIR CIRCULATED; CONSTANT TEMPERATURE AND RELATIVE HUMIDITY MAINTAINED IN EACH ROOM (*Case E*)

Case E. (Fig. 7) The temperature t_y will ordinarily be different for each room With H and M fixed, $0.24\ (t_y - t)M = H$, or

$$t_y = \frac{H}{0.24\ M} + t$$

In order to provide the proper temperature for each room, a booster coil is generally installed in each supply duct near the outlet to control the outlet temperature t_y. The amount of steam supplied to these booster units is usually controlled automatically by individual thermostats. The heat required by the booster coils depends on the temperature range through which the air is heated and the quantity of air, or

$$H_4 = 0.24\ (t_y - t_2 - t_z)M \qquad (10)$$

Total Heat to be Supplied

The total heat to be supplied (H^1) is equal to the sum of the heat requirements of the various heating units and the water heater of the washer, if any, plus the allowance for piping tax. (See preceding *Cases A* to *E*.)

Grate Area, Boiler Selection

The required grate area may be determined by the following formula:

$$G = \frac{H^1}{F \times E \times C} \tag{11}$$

where

G = required grate area, square feet.
F = calorific value of fuel, Btu per pound.
C = combustion rate, pounds per square foot of grate per hour.
E = boiler and grate efficiency, per cent.

Example 6. Using the data in Example 4, and assuming coal having a calorific value of 12,000 Btu per pound, a combustion rate of 7 lb per square foot, and a performance efficiency of 0.60, and neglecting the piping tax,

$$G = \frac{1{,}306{,}670}{12{,}000 \times 0.60 \times 7} = 26 \text{ sq ft}$$

Weight of Condensate

The normal weight of condensate to be handled from central fan systems may be estimated by means of the following formula:

$$W = \frac{60 \times Q \times \Delta t}{55.2 \times h_{fg}} \tag{12}$$

where

W = weight of condensate, pounds per hour.
Q = total volume of air, cubic feet per minute.
Δt = temperature rise of air, degrees Fahrenheit.
h_{fg} = latent heat of steam in the system, Btu per pound.

Ducts and Outlets, Air Filters, Air Washers

The design of the duct system should be based on data contained in Chapter 20. Air washers and humidifiers are described in Chapter 11. For information on air filters, see Chapter 16.

Static Pressure

The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus involved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed:

1. Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 in. of water.
2. A typical oil filter at rated capacity and velocity has a drop of 0.25 in. of water.
3. The loss of one row of a standard make tempering stack equals 0.09 in. water.
4. The loss of one row of a standard make preheater equals 0.10 in. water.
5. A standard humidifier at rated velocity may have a loss of about 0.35 in. water.
6. The loss through one row of a standard make reheater equals 0.12 in. water.
7. A fair assumption for duct losses on a simple system is 0.25 in. water.
8. The static pressure for a nozzle type outlet may be taken as 0.1 in. water.

The sum of these values equals 0.2 + 0.25 + 0.09 + 0.10 + 0.35 + 0.12 + 0.25 + 0.1 = 1.46 in. which is the static pressure against which the system must operate.

Fans and Control

The selection of fans and motors may be based on data contained in Chapter 17. Because centrifugal fans reach their maximum efficiency when working against the resistance offered by the average central fan heating system, they are well adapted to such systems and are generally used. Information on temperature control for central fan systems is given in Chapter 14.

PROBLEMS IN PRACTICE

1 ● What are the functions of (a) tempering coils, (b) preheating coils, (c) reheating coils, (d) booster coils, (e) water heaters?

a. Tempering coils raise the temperature of incoming air above the freezing point of water.

b. Preheating coils add to the air sufficient sensible heat above the dew point of the conditioned space to evaporate the amount of spray water required for humidification. They are used with humidifying type air washers.

c. Reheating coils raise the air temperature from the dew point to approximately the proper delivery temperature.

d. Booster units are used for more refined individual room temperature control.

e. Water heaters may be used in place of preheaters. The latent heat of evaporation is then supplied directly to the water.

2 ● What saving results from recirculating some of the room air and reducing the amount of outside air?

Because outside air must be heated to room temperature, reducing the amount of outside air produces a proportionate saving in heat or fuel.

3 ● What items make up the total heating load in a central fan heating system?

1. The net heat loss from the conditioned space.
2. The heat required for evaporation of water for humidification.
3. The heat required to raise the temperature of outside air to room temperature.
4. Heat losses from pipes and ducts.

4 ● Why is it necessary to determine the total static pressure of a central fan heating system?

To select a fan of maximum efficiency and to determine the power required to operate the fan.

5 ● A group of three drafting rooms, having a total volume of 27,000 cu ft, a transmission loss of 110,100 Btu per hour, and an infiltration loss of 34,200 Btu per hour on the basis of 0 F outdoors and 70 F room temperature, is to be heated by a recirculating hot blast heating system with air entering the rooms at 116 F. How many cubic feet per minute, measured at 70 F, will be required?

Substitute in Equation 3. $H = 110{,}100 + 34{,}200 = 144{,}300$ Btu per hour; $t_y = 116$ F; $t = 70$ F; $Q = \frac{55.2H}{t_y - t} = \frac{55.2 \times 144{,}300}{116 - 70} = 173{,}160$ cu ft per hour.

$\text{cfm} = \frac{173{,}160}{60} = 2886.$

6 ● In the preceding question, if the hot air loses 4 F between heater and rooms, how many pounds of steam per hour at 1-lb gage will the heating sections condense?

Substitute in Equation 12. $Q = 2886$ cfm, from solution of Question 5; $\Delta_t = 116 + 4 - 70 = 50$ F; $h_{fg} = 968$ Btu, from steam table in Chapter 1.

$$W = \frac{60 \times Q \times \Delta_t}{55.2 \times h_{fg}} = \frac{60 \times 2886 \times 50}{55.2 \times 968} = 162 \text{ lb per hour.}$$

7 ● The same rooms are converted to chemical laboratories, requiring the introduction of 12 changes of outside air, measured at 70 F, per hour to permit the exhaust fans connected to the chemical hoods to maintain only a slight negative pressure in the rooms. At what temperature must the air enter the rooms to maintain 70 F with 0 F outside?

Substitute in Equation 2. $H = 110{,}100 + 34{,}200 = 144{,}300$ Btu per hour; $Q = 12 \times 27{,}000 = 324{,}000$ Btu per hour; $t = 70$ F; $t_y = \frac{55.2H}{Q} + t = \frac{55.2 \times 144{,}300}{324{,}000} + 70 \doteq 94.6$ F.

8 ● In the preceding question, if the air drops 2 F between the heater and the rooms, how many pounds of steam per hour at 1-lb gage will the heating system condense?

Substitute in Equation 12. $Q = 5400$ cfm; $\Delta_t = 94.6 + 2 = 96.6$ F, from solution of Question 7; $h_{fg} = 968$ Btu, from steam table in Chapter 1.

$$W = \frac{60 \times Q \times \Delta_t}{55.2 \times h_{fg}} = \frac{60 \times 5400 \times 96.6}{55.2 \times 968} = 585 \text{ lb per hour.}$$

9 ● The combination hot blast heating and ventilating system for the dining rooms of a hotel is to heat the rooms to 70 F with 0 F outside, and permit the exhaust fan from the adjoining kitchen to draw 5000 cfm from the dining rooms. The transmission losses from the dining rooms total 240,000 Btu per hour. The infiltration into the dining rooms amounts to 1000 cfm from outdoors and 1000 cfm from heater rooms. How many cubic feet per minute, measured at 70 F, must be supplied the dining rooms if the air enters at 112 F?

First find the infiltration loss by substituting in Equation 1.

$t = 70$ F; $t_o = 0$; $M_o = d \times Q = 0.07495 \times 60 \times 1000 = 4497$ lb per hour. In this case d and Q are figured at 70 F. $H_o = 0.24\,(t - t_o)$; $M_o = 0.24\,(70 - 0) \times 4497 = 75{,}550$ Btu per hour.

Next by substituting in Equation 3, find the cubic feet per hour to be circulated. H = sum of transmission and infiltration losses in room $= 240{,}000 + 75{,}550 = 315{,}550$ Btu per hour; $t_y = 112$ F; $t = 70$ F; $Q = \frac{55.2H}{t_y - t} = \frac{55.2 \times 315{,}550}{112 - 70} = 414{,}700$ cu ft per hour.

cfm $= \frac{414{,}700}{60} = 6912$.

10 ● In Question 9, 3000 cfm of outside air will be drawn in by the supply fan and 3912 cfm will be recirculated. What will be the output of the heating sections in Btu per hour if there is a loss of 2 F between the heaters and the room?

The average temperature of the mixture of outdoor and recirculated air entering the heater $= \frac{3000 \times 0 + 3912 \times 70}{6912} = 39.6$ F. Air leaves the heater at $112 + 2 = 114$ F.

Referring to Equation 12, $W \times h_{fg}$ = total heat required per hour $= \frac{60 \times Q \times \Delta_t}{55.2} = H$.

$Q = 6912$ cfm; $\Delta_t = 114 - 39.6 = 74.4$ F. $H = \frac{60 \times 6912 \times 74.4}{55.2} = 558{,}900$ Btu per hour.

11 ● When the outdoor wet- and dry-bulb temperatures are 0 F, a certain printing shop is to be maintained at 75 F and 40 per cent relative humidity by means of an air conditioning system having tempering sections, an air washer, and reheating sections. The transmission loss is 80,000 Btu per hour and the infiltration is 10,000 cu ft per hour, measured at 0 F. No outside air connection is provided. How many pounds of air per hour at 120 F must be discharged to the shop?

Infiltration heat loss, by Equation 1 $= H_o = 0.24\ (t - t_o)\ M_o$. By Equation 6, $M_o = d_oQ_o = 0.08636$ (*from Table 5, Chapter 1*) $\times$ 10,000 = 863.6 lb per hour; t = 75 F; t_o = 0 F; $H_o = 0.24\ (75 - 0)\ 863.6 = 15{,}544$ Btu per hour. Total heat loss in room = 80,000 + 15,544 = 95,544 Btu per hour = H.

To secure the total weight of air to be introduced into the space, substitute in Equation 4, $M = \dfrac{H}{0.24\ (t_y - t)} = \dfrac{95{,}544}{0.24\ (120 - 75)} = 8846$ lb per hour.

12 ● In the preceding example: (a) How many Btu per hour are used to heat the room? (b) How many pounds of water must be evaporated per hour to humidify the space? (c) How many Btu will be required to evaporate this water, basing the latent heat of evaporation on the approximate figure of 1050 Btu?

a. Btu to heat room = 95,544 as derived in preceding solution.

b. Saturated air at 75 F contains 0.01877 lb of water vapor per pound of dry air. At 40 per cent relative humidity the air would contain 0.40 $\times$ 0.01877 = 0.00750 lb of water vapor per pound of dry air; at 0 F, saturated air contains 0.00078 lb of water vapor per lb of dry air. The amount of water vapor required to humidify the air = 0.00750 − 0.00078 = 0.00672 lb per cu ft. Infiltration amounts to 863.6 lb per hour as derived in the preceding solution, so 863.6 $\times$ 0.00672 = 5.80 lb of water vapor per hour required.

c. The heat required to evaporate this water = 5.80 $\times$ 1050 = 6090 Btu per hour.

Chapter 23

MECHANICAL WARM AIR FURNACE SYSTEMS

Fan Furnaces, Fans and Motors, Elimination of Noise, Air Washers and Filters, Cooling Methods, Duct Design, Controls, Selecting the Furnace, Selecting the Fan, Humidity Provision for Cooling System, Heavy Duty Fan Furnaces

MECHANICAL warm air or fan furnace heating systems, which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 24. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.
2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.
3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.
4. Humidity control is more readily attained.
5. The air may be cleaned by air washers or filters, or both.
6. Some cooling effect in summer will result from the installation of a properly designed system.
7. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.
8. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the coordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

FAN FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace

depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. Bituminous—Large combustion space with easily accessible secondary radiator or flue travel.
   b. Anthracite or coke—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is

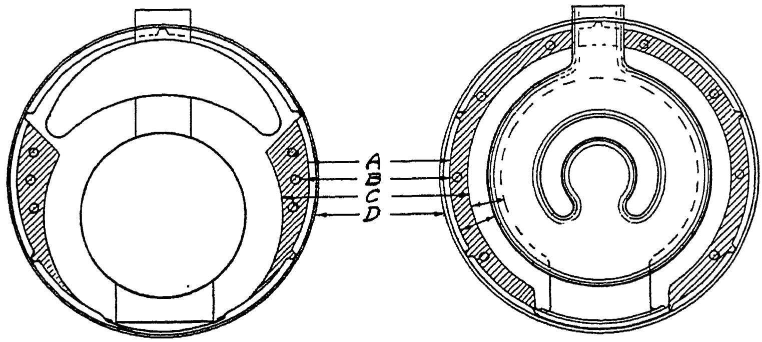

FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK
A. Liner, 1 in. from casing. *B.* Hole to vent baffle.
C. Baffle, closed top and bottom. *D.* Outer casing.

permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should

be lined with black iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2-in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be used, many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. The method of making these baffles for furnaces with top horse-shoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows the usual method of baffling square furnace casings for fan-furnace work.

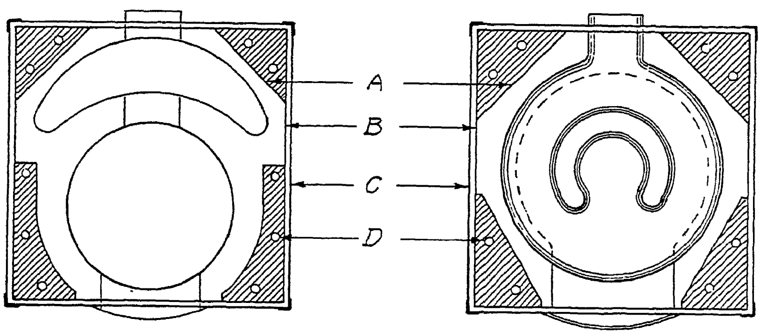

FIG. 2. METHOD OF BAFFLING SQUARE FURNACE CASING FOR FAN FURNACE WORK

A. Baffle, closed top and bottom. *B*. Liner, 1 in. from casing. *C*. Outer casing. *D*. Hole to vent baffle.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet and thus provides a larger plenum chamber. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Low tip speed is desirable for the elimination of air noise, especially where forward curved blades are used. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys

are desirable to provide a factor of safety and to allow for more rapid circulation for summer cooling.

For additional information on fans and motors, see Chapter 17.

NOISE ELIMINATION

Special attention must be given to the problem of noise elimination. The fan housing must not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings

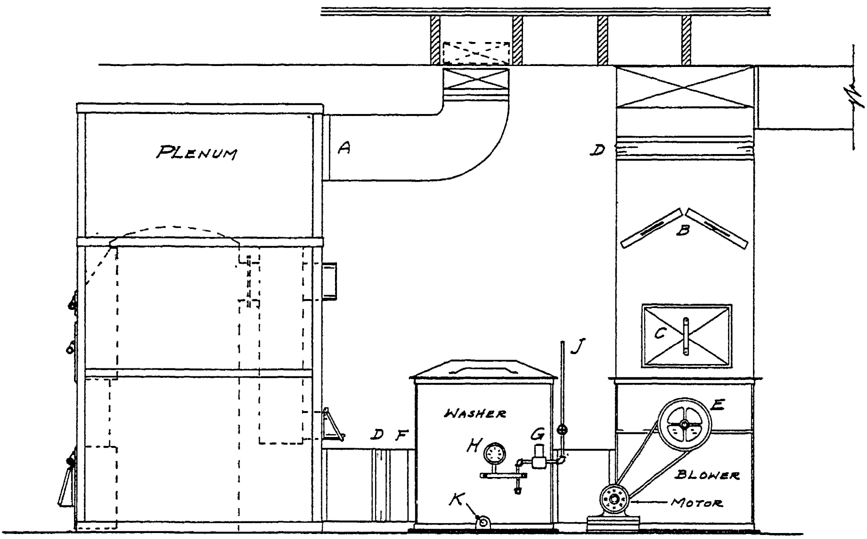

FIG. 3. COMPLETE RESIDENCE FAN FURNACE INSTALLATION SHOWING LOCATION OF FAN, FURNACE, FILTERS, PLENUM CHAMBER AND METHOD OF TAKE-OFF OF WARM AIR PIPE

A. Transition fitting.
B. Filters.
C. Capped opening.
D. Canvas connection.
E. Pulley—3 diam. V-type.
F. Eliminator.
G. Solenoid valve.
H. Pressure gage.
J. Water supply.
K. Drain.

must be carefully chosen for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with, fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. See also Chapter 18.

AIR WASHERS AND FILTERS

Washers for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Washers are usually controlled by solenoid valves wired in parallel with

the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Washers used in connection with commercial or heavy duty plants should be a regulation type of commercial washer.

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. See also Chapter 16.

COOLING METHODS

Some cooling may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the water is below the dew point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. See also Chapters 9 and 10.

Study of these problems sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating Association* is in progress at the University of Illinois. The following conclusions may be drawn from the studies thus far completed, subject to the limitations of the conditions under which the tests were run[1]:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

[1]See A.S.H.V.E. research paper entitled Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A. S. H. V. E. Journal Section *Heating, Piping and Air Conditioning*, February, 1933).

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The results of the tests suggest the use of a fan at night either to provide more comfortable conditions during the following day without provision for cooling, or to reduce the load required for cooling during the following day. Experience has shown that the volume of air required for cooling, depending upon the climate and the construction of the building, must usually be from 50 to 100 per cent greater than that required for heating. If the size of the fan is based upon the summer requirement, its output may be reduced sufficiently to meet winter heating needs.

7. Attic exhaust fans are becoming popular adjuncts for night duty. (See Chapter 13.)

DUCT DESIGN

The ducts may be either round or rectangular. Rectangular ducts should be as nearly square as possible; the width should not be greater than four times the breadth. The radii of elbows should be not less than

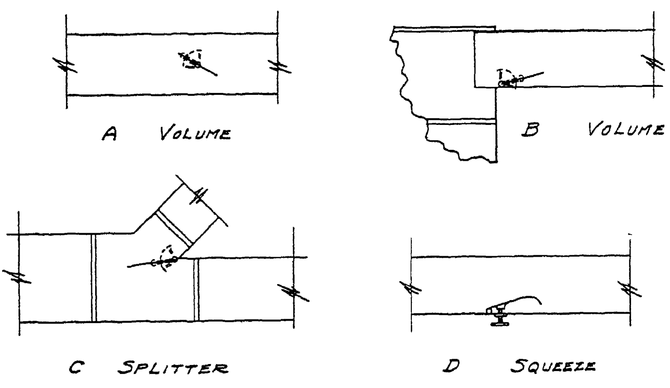

FIG. 4. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

The ducts or piping may be designed either as a trunk line system or as a system of individual ducts from the furnace casing to each register. The engineering problems incident to the design of a trunk line system are somewhat more difficult than for the individual duct system. The trunk line system is generally a tailor-made job, whereas the individual duct system with which either round or square ducts may be used may frequently be assembled from stock materials and thus installed at a considerable saving. Individual ducts may frequently be grouped to simulate

a trunk duct system in appearance. The design of ducts for air flow is described in Chapter 20.

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely in balance without the use of dampers. Special care must be used in the design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. See Figs. 1 and 2, Chapter 20.

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 4.) *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 4.) *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 4.) It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Supply and Return Air Registers

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Unless registers located in the baseboard are well proportioned and designed to harmonize with the trim, they may be unsightly. Registers which are located in side walls above the baseboard or in the ceiling should be of an effective air-diffusing type. All registers should be sealed against leakage around the borders or margins.

Velocities through registers may be reduced by the use of registers

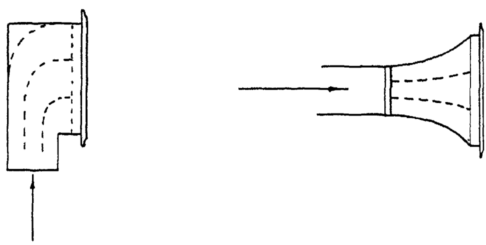

FIG. 5. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated in Fig. 5. Merely to use a larger register may not result in materially reduced velocities unless such diffusers are used.

Care should be exercised in making the connection between the supply register and its box to prevent streaking of the wall. All warm air registers should be equipped with dampers or, better, with diffuser dampers which may be used to direct air currents in such a way that they will not be objectionable. (See Chapter 19.)

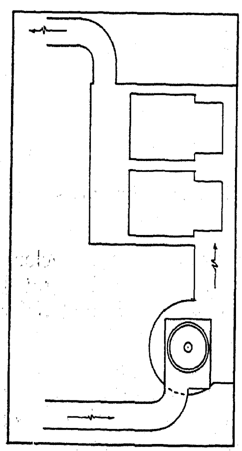

FIG. 6. HEATER ARRANGED FOR COMPLETE RECIRCULATION OF AIR

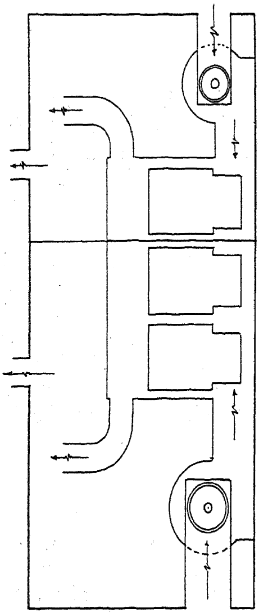

FIG. 7. TWO BATTERIES OF HEATERS AND FANS FOR INDEPENDENT SERVICE USING OUTSIDE AIR AND EXHAUSTING TO ATMOSPHERE

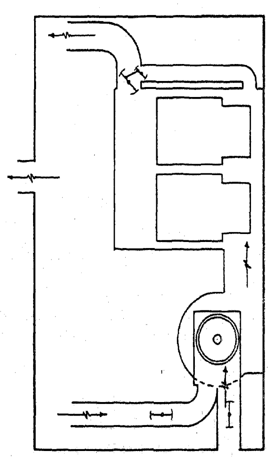

FIG. 8. HEATER ARRANGED FOR PARTIAL RECIRCULATION, ALSO SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR CHAMBERS, AND PARTIAL EXHAUST TO ATMOSPHERE

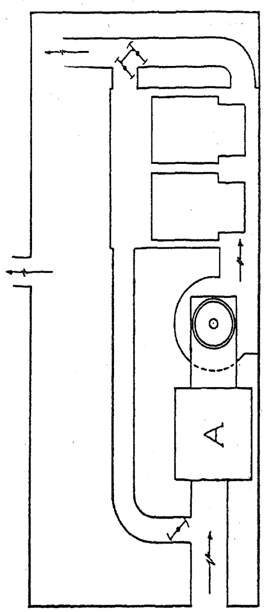

FIG. 9. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in the system can be largely eliminated through proper care in the planning and installation of the control system. The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.
3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.
4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To prevent cold air delivery when heat supply is insufficient.
6. To avoid heat loss through the chimney by keeping stack temperatures low.
7. To provide quick response to the thermostat. with protection against overrun.
8. To provide for humidity control.
9. To provide a means of summer control of cooling.
10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.
2. A *furnacestat* located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the furnacestat is in the main duct near the frame opening from the bonnet.
3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 225 F.
4. On oil and gas burner installations, a control is usually included which will shut down the system if the fire goes out or if there is a failure of the ignition system.
5. A *humidistat* to regulate the moisture supplied to the rooms.
6. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying.

While it is usually all right to start and stop the fan in residential control work and in auditoriums and other places where many people may gather, the fan should as a rule be allowed to run continuously and the control should be cared for in other ways.

SELECTING THE FURNACE

The following formula may be used to compute the grate area of a residence furnace, assuming a ratio of heating surface to grate area of 20 to 1:

$$G = \frac{H}{F \times C \times E} \tag{1}$$

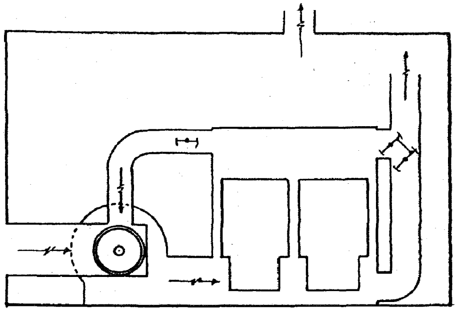

Fig. 10. Heater Arranged to Use Complete Outside Air with Tempered Air Chamber above Warm Air Chamber Showing Mixing Damper to Temper Air, Also Mixing Damper to Duct from Warm and Tempered Air Chambers

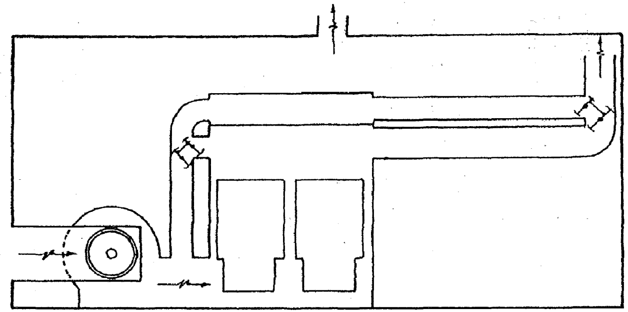

Fig. 11. Heater Arranged to Use Outside Air, with Tempered Air Chamber above Warm Air Chamber Showing Double Duct Distributing System and Mixing Dampers at Base of Risers

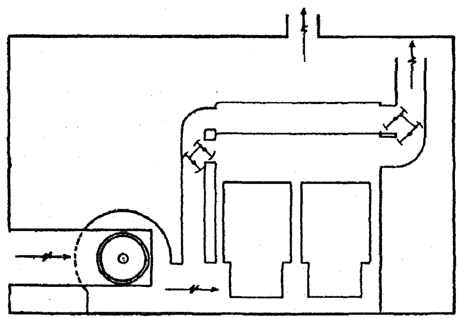

Fig. 12. Heater Arranged for Outside Air with Warm Air Connection to Fan Inlet to Temper Air, also Mixing Dampers from Warm Air and Tempered Air Chamber

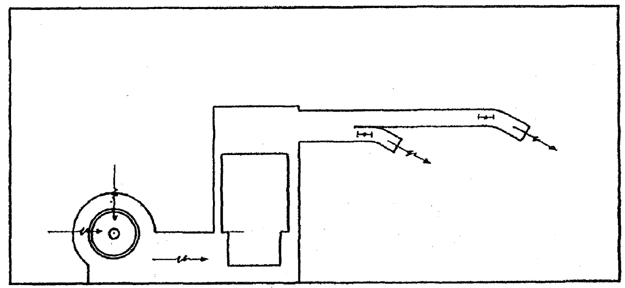

Fig. 13. Heater Arranged for Recirculation with Trunk Line Distributing System Showing Temperature Control by Air Volume at Outlet, or Direct on Heater

where

G = required grate area, square feet.
H = total heat loss from building, Btu per hour.
F = calorific value of coal, Btu per pound.
C = combustion rate in pounds of fuel per square foot of grate per hour.
E = furnace efficiency based on heat available at register faces.

In practice it is customary to use the following constants:

F = 13,000 (For specific values, see Table 1, Chapter 27).
C = 5 to 10 lb (Use 8 lb as maximum in residence work).
E = 55 per cent to 65 per cent depending on fuel burned. Lower efficiency must be used with highly volatile solid fuel.

Where ratio of heating surface to grate area is less or greater than 20 to 1, deduct or add 2 per cent from or to rating of furnace for each unit decrease or increase in ratio, as the case may be. The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick. Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturers' Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula:

$$R = \frac{H}{0.9} \tag{2}$$

where

H = total heat loss from building in Btu per hour.
R = official *A.G.A.* output rating of the furnace in Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.56H \tag{3}$$

where

I = Btu per hour input.

The factor 1.56 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

SELECTING THE FAN

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the following items:

1. The frictional resistance of a warm air trunk or leader.
2. The frictional resistance of a return air trunk or duct.
3. The resistance to the flow of total volume of air through the furnace casing or hood, which is usually considered from 0.10 to 0.15 inches of water.
4. The frictional resistance through any other accessories, such as washers or filters.
5. A factor of safety of 10 per cent of the resistance calculated above.

HUMIDITY

Mechanical warm air systems offer an excellent means of proportioning and distributing moisture-bearing air; consequently, during the winter months humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the limits of the generally accepted standards. See Chapters 2 and 3 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface.

Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, a trend in present practice toward heating the water in addition to heating the air. Equipment for doing this may make use of sprays, or it may take the form of water circulating coils placed within the combustion chamber and connected by pipes to the humidifier pans where a constant water level is maintained by some separate float device. (See Chapter 11.)

PROVISION FOR COOLING SYSTEM

If the system is to be used for cooling, the following provisions should be made:

1. Where cooling is to be secured through air circulation only:
   a. Provide for an increase of 50 to 100 per cent in fan capacity through multi-speed pulleys or other means.
   b. If basement air or outside night air is to be used, provide suitable basement opening in duct system, or outdoor air intake.
2. Where water *below* 55 F or artificial refrigeration or ice is to be used:
   a. Provide outside air duct for circulation of cool night air for economy.
   b. Make provision in return duct system for cooling unit.
   c. Make provision for control of the fan speed, during winter operation, to give a sufficient and draftless air movement.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. A few possible arrangements are shown in Figs. 6 to 13, inclusive.

Most manufacturers of heavy duty furnaces rate their furnaces in Btu per hour and also in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area on furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

Control of temperature is secured through (1) controlling the quantity of heated air entering the room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 22. Ducts are designed by the method outlined in Chapter 20.

PROBLEMS IN PRACTICE

1 ● Why do furnaces designed to burn bituminous coal, oil, or gas require larger combustion spaces than those designed for anthracite?

Anthracite burns largely as fixed carbon whereas gas and oil burn as gases, and as much as 50 per cent of bituminous coal burns as a gas. Ample space must be provided for the intimate mixture of these gases with the oxygen of the air to secure proper combustion.

2 ● A furnace has the following dimensions: Grate diameter, 24 in.; casing diameter for gravity air flow, 56 in.; combustion chamber diameter, 30 in. What is the unobstructed area required for passage of air across the heating surface when a motor-driven blower, operating at an outlet velocity of 1200 fpm, delivers 1600 cfm into the casing near its bottom?

For residence applications using small blowers, an air outlet velocity of about one third of the blower outlet velocity is considered good practice.

$$\text{Air-pass velocity} = \frac{1200}{3} = 400 \text{ fpm.}$$

$$\text{Air-pass area} = \frac{1600}{4} = 4 \text{ sq ft} = 576 \text{ sq in.}$$

3 ● In Question 2 what would be the gap between the chamber and the baffle when the chamber is centered in the casing?

Area of combustion chamber (30-in. diam)	706.9 sq in.
Area of air pass	576.0 sq in.
Total area	1282.9 sq in.

The diameter of a circle with an area of 1282.9 sq in. is 40.4 in. One half of the difference between the diameters is the amount of gap.

$$\text{Gap} = \frac{40.4 - 30.0}{2} = 5.2 \text{ in.} = \text{approximately } 5\tfrac{1}{4} \text{ in.}$$

4 ● Why should secondary surface be designed for easy cleaning?

If the combustion is not perfect, soot is formed immediately above the fire and is apt to form a deposit on the secondary surface from which it should be removed. If the secondary surface is so designed that there are horizontal passages, fine gray ash will settle out in these to form an insulation between the hot gases of combustion and the metal of the furnace; consequently, these should be readily cleaned. If the passages are vertical they are largely self-cleaning of ash, but provision should be made for easy and thorough cleaning of the collection chamber below them.

5 ● Why is baffling inside the casing necessary on fan systems?

Because the movement of air is independent of its temperature, air must be guided by baffles of one form or another to bring it in contact with the hot surfaces so it will not pass through the casing unheated. On the other hand, if the air is held against a hot surface too long it might become overheated, for the average register temperature on a fan system should not exceed 120 F.

6 ● Why do buildings which are intermittently used require more heating capacity than buildings constantly used?

Between heating periods the intermittently used building is allowed to cool down. All of the material in the building loses heat, and before the building can be reheated to a comfortable temperature this material must also be reheated.

7 ● What practical points should be observed in designing a fan system in order to eliminate noise?

a. Use a large fan so it can be run at slow speed.

b. Set the fan and motor on a solid foundation.

c. Insulate the fan and motor from the foundation with rubber, cork, or other springy material according to the principles given in Chapter 18, provided, of course, that such insulation is of value.

d. See that the air velocity is not too high in the ducts. Properly designed splitters in the elbows will avoid high velocities at the turns in cases where the velocity through the ducts themselves is not too high.

e. Use canvas connections between the ducts and any running equipment.

f. Be sure the ducts have a relatively smooth interior and are rigid.

Chapter 24

GRAVITY WARM AIR FURNACE SYSTEMS

Procedure for Design, Estimating Heating Requirements, Sizes of Leader Pipes, Proportioning Wall Stacks, Register Sizes, Recirculating Ducts and Grilles, Return Connection to Furnace, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 23. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

PROCEDURE FOR DESIGN

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building.
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers.
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

[1]All figures and much of the engineering data which follow are from *Bulletins No.* 141, 188 and 189, Warm Air Furnaces and Heating Systems, Part II, by Professor A. C. Willard, A. P. Kratz, and V. S. Day, Engineering Experiment Station, University of Illinois.

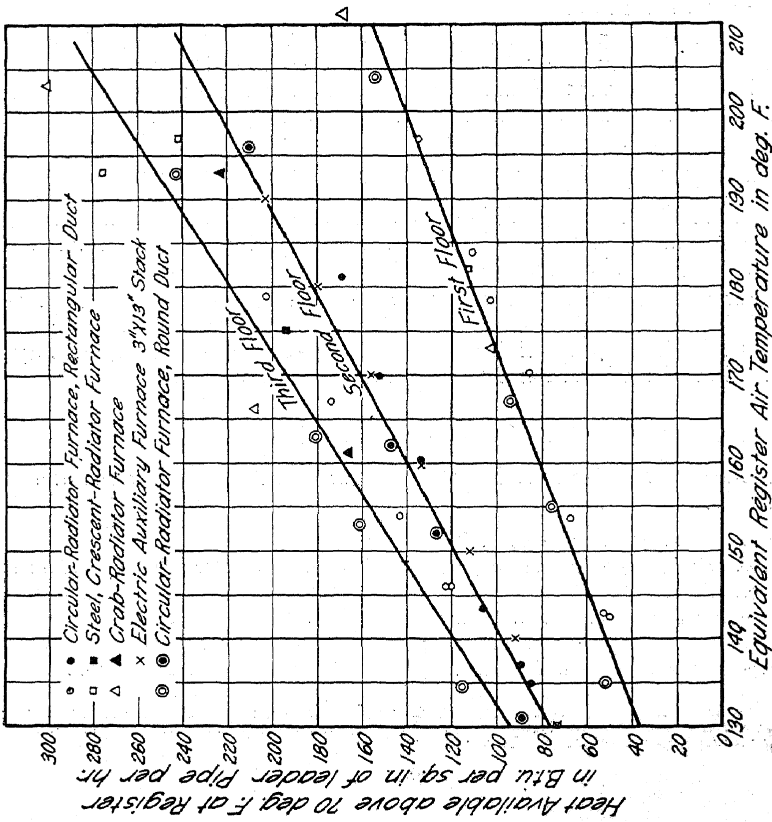

FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

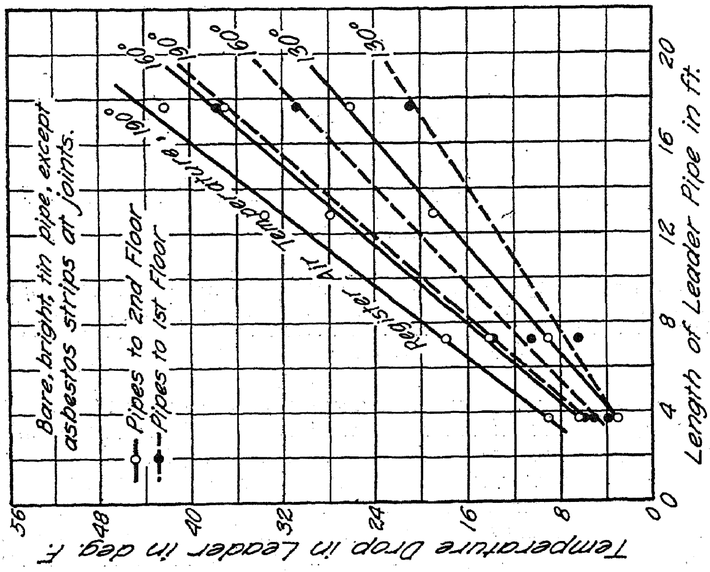

FIG. 2. INFLUENCE OF LEADER PIPE LENGTH ON TEMPERATURE LOSS IN AIR FLOWING THROUGH PIPE

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This *size* should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 7, taking into consideration the transmission losses as well as the infiltration losses.

SIZES OF LEADER PIPES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \qquad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \qquad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \qquad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \qquad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \qquad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be very thoroughly covered or else the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

While Fig. 1 takes care of the drop in temperature in straight leaders up to 8 ft in length connected to stacks having about 75 per cent of the

area of the leader, the designer must make allowances for all other conditions. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. It is not considered good commercial practice to specify diameters except

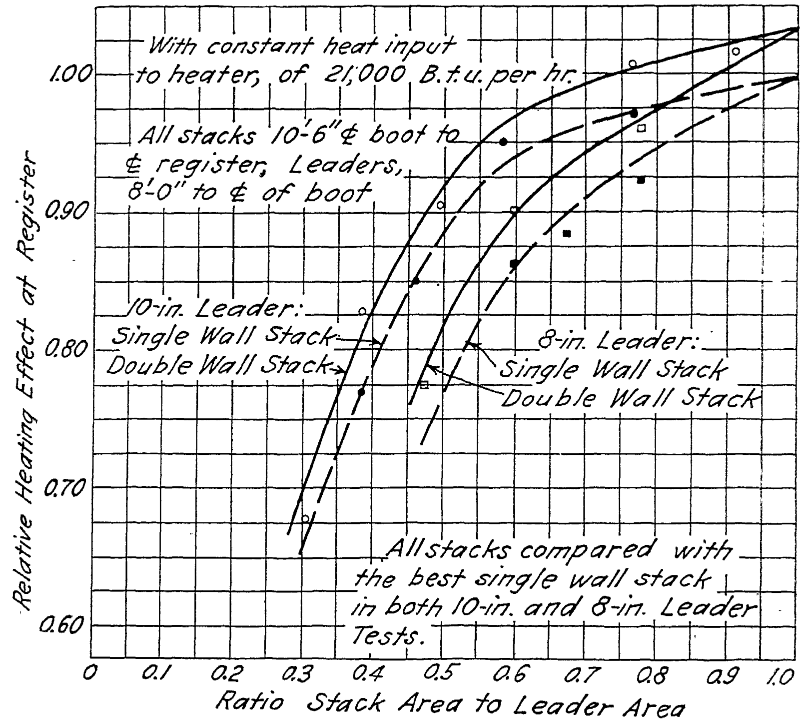

FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

Note.—Exterior surface of all ducts is bright tin except at joints where asbestos sealing strips are used.

in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length are to be avoided or should receive very special attention.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader which has been selected from Fig. 1. So long as the leader is short and straight as was the case for Fig. 1,

such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not very serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this stack to leader area ratio is a very important consideration. Specific data for a great variety of cases are presented in Figs. 4 and 5 and the designer should check the stack to leader combinations with the nearest comparable case as shown in these figures. Any second-floor stack supplying heat to a room whose heat loss is 9,000 Btu or more (see Figs. 4 and 5 which show that high temperatures are necessary if rooms of more than 9,000 Btu requirement are heated by one stack each in 4-in. studding) should be run within 6-in. studded walls or should have multiple stacks. Stack sections, wherever possible, should be changed from the thin rectangular to the more nearly square shape.

REGISTER SIZES

The registers used for discharging warm air into the rooms should have free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First-floor registers may be of the baseboard or floor type, with the former location preferred.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, in excess of the total area of warm-air pipes, and at all points where the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return must be placed to effectively receive the cold air returning by way of the stairs.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made

oversize and favored in every way. This precaution is particularly important when long ducts and short ducts are used in the same system.

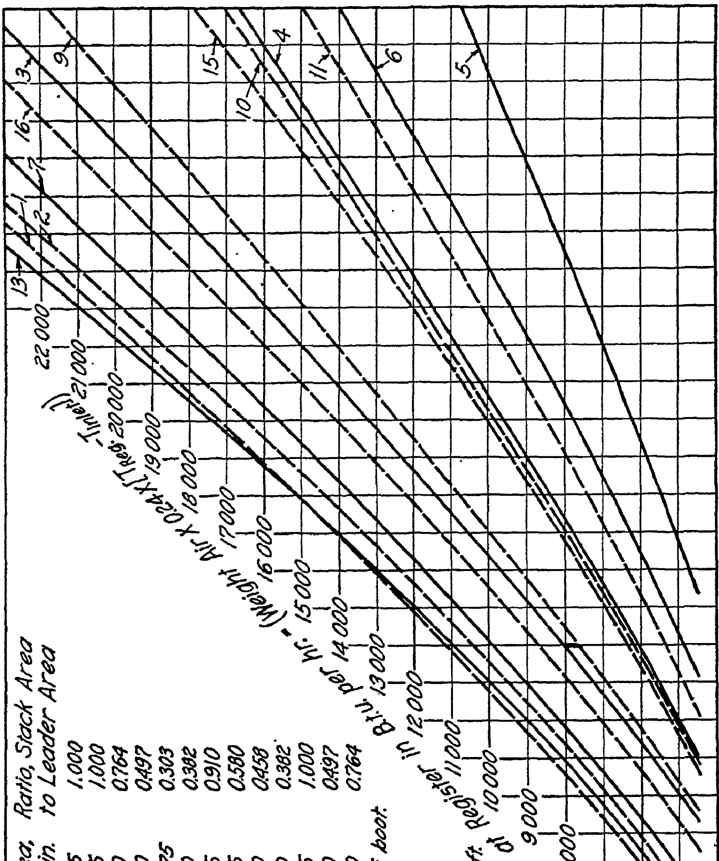

FIG. 4. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces closed off or badly exposed. Metal linings are advisable in such ducts. It is important that these ducts be free from unnecessary

friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

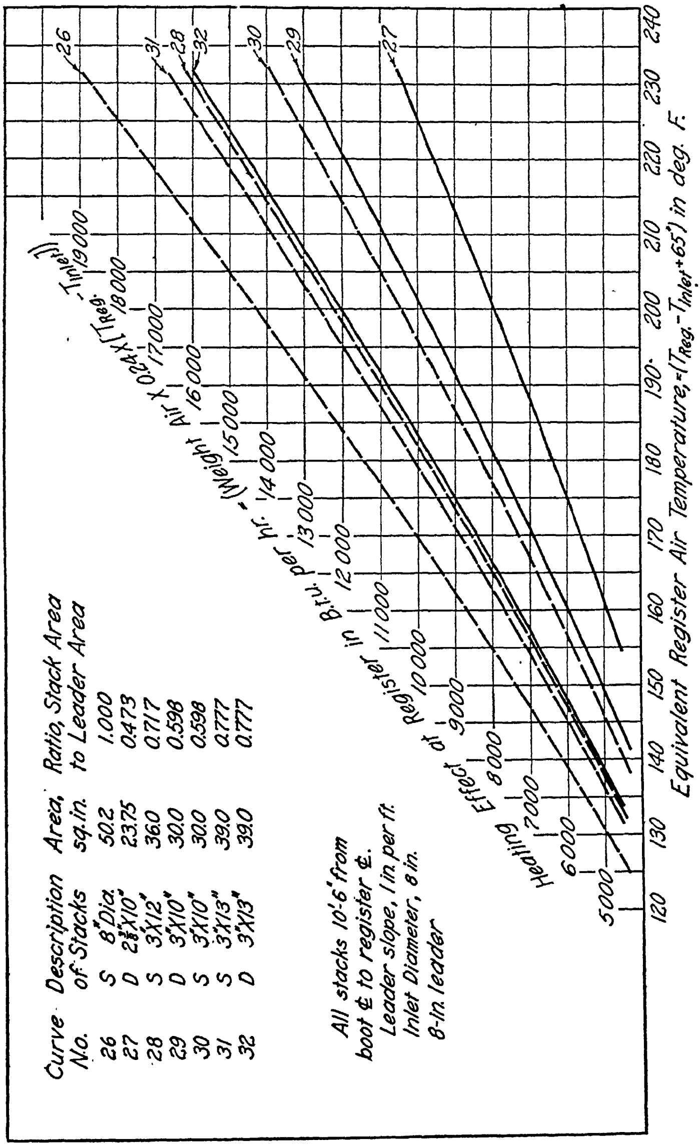

FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

Return Connection to Furnace

Circulation is accelerated if the drop to the furnace is through a round inclined pipe with, say, two 45-deg elbows rather than through a vertical drop and two 90-deg elbows. The top of the shoe should never enter

the casing above the level of the grate in the furnace. To accomplish this the shoe must be wide.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.

2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.

3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

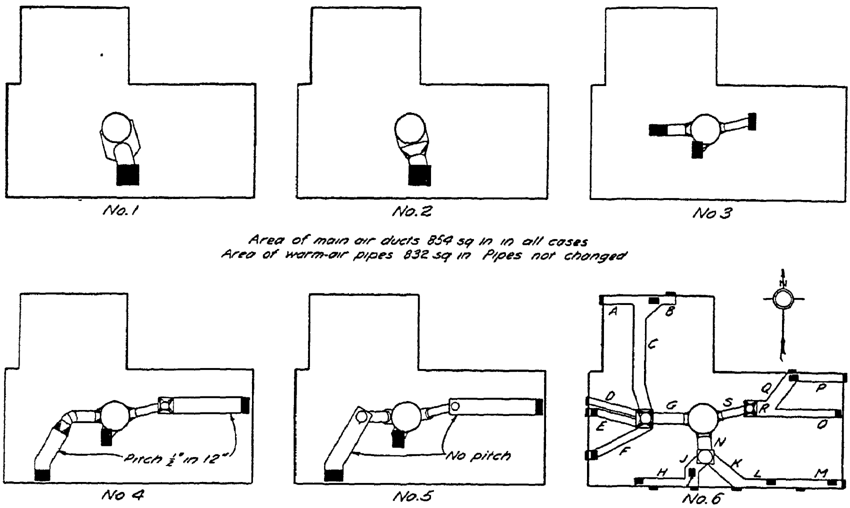

Fig. 6. Arrangement of Cold Air Returns for Six Installations

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is easily obtained by finding the sum of the leader pipe areas as already designated.

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken as 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by reference to a set of typical furnace performance curves (Fig. 7) for a cast-iron circular radiator furnace with a 23-in. diameter grate and 50-in. diameter casing. The conditions shown on these curves which seem to

[2]Investigation of Warm-Air Furnaces and Heating Systems, Part *N*, by A. C. Willard, A. P. Kratz and V. S. Day (University of Illinois *Engineering Experiment Station Bulletin No.* 189).

approximate nearest to the 175 F register warm-air temperature are: combustion rate, 7 lb; warm-air register temperature, 173 F; efficiency of the furnace, 58.5 per cent. The third factor is efficiency, which, in turn, is a function of the combustion rate varying with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

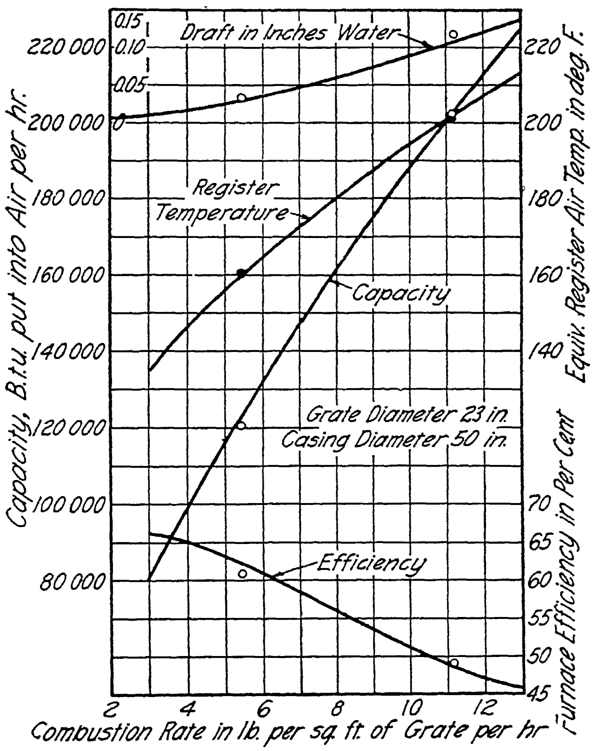

Fig. 7. Typical Performance Curves for a Warm Air Furnace and Installation in a Three Story Ten Leader Plant, Operating on Recirculated Air

From the relation existing among these factors it is found (Fig. 7) that the capacity of the furnace under test is 147,750 Btu per hour for the total grate, which gives the capacity at the furnace bonnet per square foot of grate as 51,200 Btu and per square inch of grate as 356 Btu per hour.

Suppose it is desired to select a furnace to deliver air to the rooms at a register temperature approximating 160 F rather than 175 F. Referring to the curves, the relation is: combustion rate, 5.5 lb; register warm-air temperature, 160 F; and efficiency of the furnace, 62 per cent. Under this condition the capacity of the furnace at the furnace bonnet per square foot of grate area is 43,200 Btu per hour, and per square inch of grate it is 300 Btu per hour. From these performance values, the grate area for any

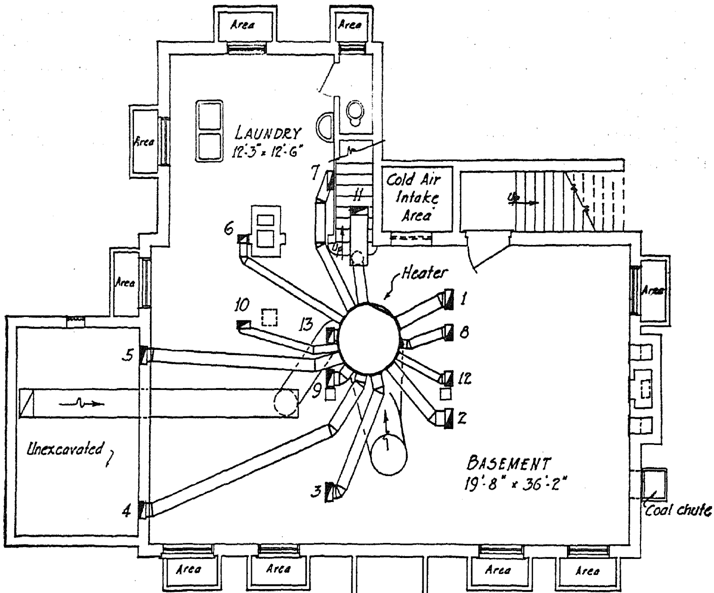

FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

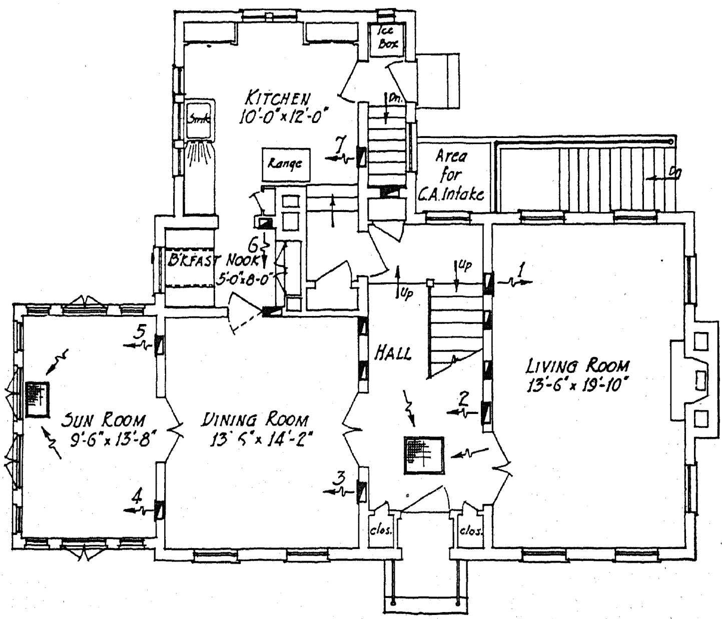

FIG. 9. FIRST-FLOOR PLAN, RESEARCH RESIDENCE

plant requirement (allowing 20 per cent heat loss between furnace and registers) will be:

$$\text{Grate area (175 F register temperature), square inches} = \frac{1.2\ H}{356} = 0.0034H \quad (7)$$

$$\text{Grate area (160 F), square inches} = \frac{1.2\ H}{300} = 0.0040H \quad (8)$$

Here H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the fresh air, if any, at intake. This fresh air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have a fresh air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 20 per cent loss between furnace and registers.

It is not always possible to obtain performance curves, and the following method is suggested as being a close check. An addition of 2 per cent of the furnace capacity is proposed for each unit that the heating surface to grate area ratio of the furnace exceeds 20. This addition is based on tests made at the University of Illinois, of four types of furnaces having various ratios of heating surface to grate area.

Let E = efficiency of the furnace.
f = fuel value of the coal, Btu per pound.
p = pounds of coal burned per square foot of grate per hour.
R = ratio of heating surface to grate area.
H = total heat requirements of the house.

$$\text{Grate area, square inches} = \frac{1.2 \times 144\ H}{Efp\ [1 + 0.02\ (R - 20)]} \text{ for all inside air.} \quad (9)$$

For coal having a heat value of 12,000 Btu, and a furnace having 60 per cent efficiency, with 6 lb of coal burned per square foot of grate per hour, and 20 sq ft of heating surface for 1 sq ft of grate, this becomes:

$$\text{Grate area, square inches} = \frac{1.2 \times 144\ H}{0.60 \times 12{,}000 \times 6} \text{ for all inside air,} \quad (10)$$

and for another furnace having 24 sq ft of heating surface for 1 sq ft of grate the expression is

$$\text{Grate area, square inches} = \frac{1.2 \times 144\ H}{0.60 \times 12{,}000 \times 6\ [1 + 0.02\ (24 - 20)]} \quad (11)$$

The air temperatures at the registers corresponding to the conditions of Equation 11 would be approximately 165 F, and for 175 F and 12,000 Btu the combustion rate would be about 7.5 lb with an efficiency of 57 per cent, using the curves of Fig. 7 as a guide.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

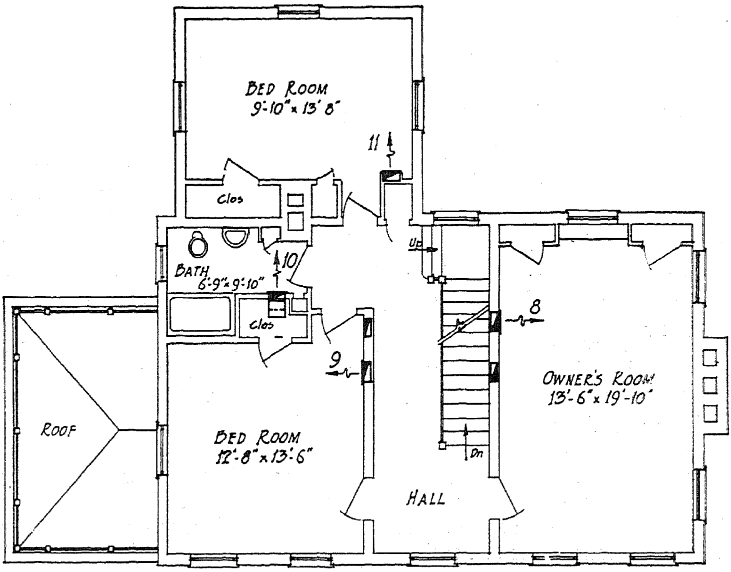

FIG. 10. SECOND-FLOOR PLAN, RESEARCH RESIDENCE

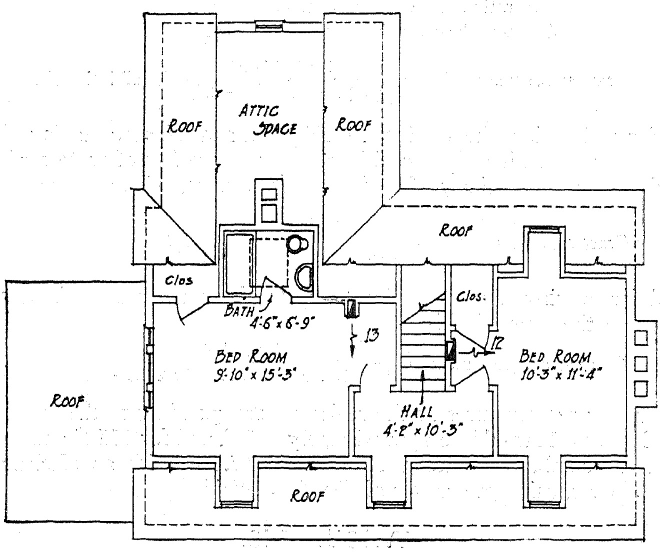

FIG. 11. THIRD-FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating Association* erected at the University of Illinois[3].

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in. leader area. See summary, Table 1; also example under Standard Code[4], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 in. × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See Summary Table; also example under Standard Code[4], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 11.4, say 12 in.

Stack area = 0.7 × 90 = 63 sq in. = say 5 in. × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 in. × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code [4] Method. See Art. 3, Sec. 1, 2, 3.)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12} + \frac{405}{60} + \frac{2405}{800}\right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12} + \frac{394}{60} + \frac{2275}{800}\right) 6 = 90 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate area = 0.0034 × 132,370 = 450 sq in. = 24 in. diameter at 175 F register temperature. (Equation 7)

Grate area = 0.0040 × 132,370 = 530 sq in. = 26 in. diameter at 160 F register temperature. (Equation 8)

If provision should be made for certain outside air circulation, then increase the building heat loss by, say 25 per cent and obtain by Equation 7 a 27-in. grate and by Equations 8 and 10 a 29-in. grate.

Experiments at the University of Illinois[5] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan,

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[3]Plans used with permission. Bathroom on third floor not heated.

[4]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the eighth edition of which may be obtained from the *National Warm Air Heating Association*, 3440 A.I.U. Building, Columbus, Ohio.

[5]See University of Illinois *Eng. Exp. Sta. Bulletin No.* 120, p. 129.

with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft.*

Table 1. Summary of Data Applied to Warm Air Research Residence

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq In.	Stack Area Sq In. $0.7 \times LA$	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		$= 0.009H$				
Living	17250	155	....	14		14×16
Dining	6810	61	....	9		8×12
Breakfast	2300	21	....	8		8×10
Kitchen	9210	83	....	11 or 12		12×14
Sun	25710	230	....	Two 12		Two 12×14
Hall and stair	12570	113	....	12		12×14
Second Floor		$= 0.006H$				
Owner's	15030	90	63	11 or 12	5×12	12×14
S. W. Bed	9800	59	41	9	$3\frac{1}{2} \times 12$	8×12
Bath	2450	15	10	8	3×10	8×10
N. Bed	14800	89	62	11 or 12	5×12	12×14
Third Floor		$= 0.005H$				
E. Bed	8220	41	29	8	3×10	8×10
W. Bed	8220	41	29	8	3×10	8×10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[6].

[6]See University of Illinois *Eng. Exp. Sta. Bulletin No.* 141, p. 79.

PROBLEMS IN PRACTICE

1 ● A first story dining room has a calculated heat loss of 12,000 Btu per hour.
a. What size leader pipe should be used for 175 F register air temperature?
b. What size register?

a. Leader area $= \frac{12,000}{111} = 108.1$ sq in. Use leader with diameter of 12 in.

b. Register gross area $= \frac{108}{0.7} = 154$ sq in. Use 12 in. by 14 in. register.

2 ● A third-story bedroom has a calculated heat loss of 12,000 Btu per hour.
a. What size leader pipe should be used for a 175 F register air temperature?
b. What size stack?
c. What size register?

a. Leader area $= \dfrac{12{,}000}{200} = 60$ sq in. Use leader with diameter of 9 in.
b. Stack area $= 0.7 \times 60 = 42$ sq in. Use stack 3½ in. by 12 in.
c. Register gross area $= \dfrac{60}{0.7} = 85.7$ sq in. Use register 8 in. by 12 in.

3 ● The calculated heat loss of a house is 130,000 Btu per hour. Find the grate area required for the furnace under the following conditions:

Heating value of coal = 12,500 Btu per lb.
Furnace efficiency = 55 per cent.
Combustion rate = 7.5 lb per sq ft per hour.
Ratio of heating surface to grate area of furnace = 20 to 1.
Register temperature = 175 F.
Loss between furnace and registers = 20 per cent.

$$\text{Grate area} = \frac{1.2 \times 144 \times 130{,}000}{0.60 \times 12{,}500 \times 7.5} = 399.5 \text{ sq in.}$$

Grate diameter = 22.6 in.
Use grate with diameter of 23 in.

4 ● If in Question 3 the conditions were the same except that the ratio of heating surface to grate area of furnace was 24 to 1, what size grate would be required for the furnace?

$$\text{Grate area} = \frac{1.2 \times 144 \times 130{,}000}{0.60 \times 12{,}500 \times 7.5} \times \frac{1}{1 + 0.02\,(24 - 20)} = \frac{399.5}{1.08} = 370 \text{ sq in.}$$

Grate diameter = 21.7 sq in.
Select grate with diameter of 22 in.

5 ● Name the items involved in the design of a furnace heating system.

a. Heat loss from each room, Btu.
b. Area and dimensions of warm-air pipes in basement, inches.
c. Area and dimensions of vertical pipes, inches.
d. Free and gross area and dimensions of warm-air registers, inches.
e. Area and dimensions of recirculating or outside air ducts, inches.
f. Free and gross area and dimensions of recirculating registers, inches.
g. Size of furnace necessary to supply the warm air to overcome the heat loss.
h. Area and dimension of chimney and smoke pipe, inches.

6 ● Discuss the design features of recirculating ducts.

a. Their area should be equal to or greater than that of the supply ducts.
b. They should be streamlined, and have a minimum number of turns.
c. All runs should be as short as possible.
d. Account should be taken of all cold walls and window areas in determining sizes and positions of return air inlets.
e. The return line should be pitched downward toward the furnace. It should be designed to minimize friction.
f. The top of the shoe or boot should never be above the grate level.

7 ● Discuss the use of a booster fan. What effect has a booster fan at low operating temperatures? At high ones?

A booster fan is useful in accelerating the air flow past the surface of a low temperature furnace, where only a small weight differential in the air is created, and in unbalancing a gravity system so flow is established. The first use involves the entire plant, and increases efficiency about 10 per cent with low temperature operation; the second involves only the leaders in which air flow is accelerated. At high operating temperatures the difference in weight between warm outgoing air and cool incoming air is great enough to make a booster unnecessary with ordinary gravity systems.

Chapter 25

BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output, Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section,* and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers may be of round pattern with circular grate and horizontal pancake sections joined by push nipples and tie rods, or of rectangular pattern with vertical sections. The latter type may be either of outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers, or assembled with push nipples and tie rods, in which case the water and steam connections are internal.

Cast-iron boilers usually are shipped knocked down to facilitate handling at the place of installation where assembly is made. One of the chief advantages of cast-iron boilers is that the separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature is of importance in making repairs to or replacing a damaged or worn out boiler and should be given consideration in the original selection. Sufficient space should be provided in the boiler room for assembling the boiler and for disassembling it conveniently if repairs are needed. With the outside header type of boiler a damaged section in the middle of the boiler can be removed without disturbing the other sections and sufficient side clearance should be provided for this contingency.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of steam radiation. For larger loads, cast-iron boilers must be installed in multiple, or a steel boiler must be used. In most cases cast-iron boilers are limited to working pressures of 15 lb for steam and 30 lb for water. Special types are built for hot water supply which will withstand higher local water pressures.

STEEL BOILERS

Two general classifications may be applied to steel boilers: *first,* with regard to the relative position of water and hot gases, distinguished as fire

tube or water tube; *second*, with regard to arrangement of furnace and flues, as (1) horizontal return tubular (HRT) boilers, (2) portable (self-contained) firebox boilers with either water or fire tubes, and (3) water tube boilers of the power type.

Fire tube boilers are constructed so that the water available to produce steam is contained in comparatively large bodies distributed outside of the boiler tubes, the hot gases passing within the tubes. In *water tube* boilers, the water is circulated within the boiler tubes, heat being applied externally to them.

The *HRT boiler* is the oldest type and consists of a horizontal cylindrical shell with fire tubes, enclosed in brickwork to form the furnace and

TABLE 1. PRACTICAL COMBUSTION RATES FOR SMALL COAL-FIRED HEATING BOILERS OPERATING ON NATURAL DRAFT OF FROM ¼ IN. TO ½ IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

combustion chamber. All heating surfaces and the interior of the boiler are accessible for both cleaning and inspection. Horizontal return tubular boilers, especially the larger sizes, should be suspended from structural columns and beams independent of the brick setting. Small HRT boilers sometimes are supported by brackets resting on the brick setting.

Portable firebox boilers are the more generally used type of steel heating boilers, their outstanding characteristic being the water-jacketed firebox which eliminates virtually all brickwork. They are shipped in one piece from the factory and come to the job ready for immediate hook-up to piping. They may be of welded or riveted construction and have either water or fire tubes. Manufacturers' catalogs usually list heating surface as well as grate area. The elimination of brickwork also makes this type the most compact of steel boilers as well as the lowest in first cost.

Water tube boilers. For large heating loads water tube boilers are quite frequently used. They usually require more head room than other types of boilers but require considerably less floor space and make possible a

much higher rate of evaporation per square foot of heating surface, with proper setting, baffling and draft. Water tube boilers used for heating purposes are brick set, supported on structural steel columns and have the brick setting encased in an insulated steel housing to prevent air infiltration and to minimize heat losses. For large heating loads at a high rate of evaporation, such boilers should be operated at pressures above 15 lb per square inch with a pressure-reducing valve on the connection to the heating main.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to suit the particular burner used, have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal fired boiler converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or

where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 28.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. Investigations[1] by the U. S. Bureau of Mines show that:

1. A boiler in which the heating surface is arranged to give long gas passages of small cross-section will be more efficient than a boiler in which the gas passages are short and of larger cross-section.
2. The efficiency of a water tube boiler increases as the free area between individual tubes decreases and as the length of the gas pass increases.
3. By inserting baffles so that the heating surface is arranged in series with respect to the gas flow, the boiler efficiency will be increased.

The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about

[1]See *U. S. Bureau of Mines Bulletin No.* 18, The Transmission of Heat into Steam Boilers.

3300 Btu per sq ft per hour for hand-fired boilers and 4000 Btu per sq ft per hour for mechanically fired boilers when operating at *design load*[2]. When operating at *maximum load*[2] these values will run between 5000 and 6000 Btu per sq ft per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted three solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[3], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[3] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[4]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[5] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers. The *Steel Heating Boiler Institute* suggests a single number dimensional rating in the *S.H.B.I.* Code for the Rating of Low-Pressure Heating Boilers by Their Physical Characteristics[6].

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of

[2]For definitions of design load and maximum load see pages 411 and 412.

[3]See A.S.H.V.E. Transactions, Vol. 35, 1929. Also Chapter 41.

[4]See A.S.H.V.E. Transactions, Vol. 36, 1930. Also Chapter 41.

[5]See A.S.H.V.E. Transactions, Vol. 37, 1931. Also Chapter 41.

[6]See Rating of Heating Boilers by Their Physical Characteristics, by C. E. Bronson (A.S.H.V.E. Transactions, Vol. 36, 1930).

boiler horsepower instead of in Btu per hour or square feet of equivalent radiation.

Boiler Horsepower: The evaporation of 34.5 lb of water per hour *from and at 212 F* which is equivalent to a heat output of $970.2 \times 34.5 =$ 33,471.9 Btu per hour.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at this same temperature and at atmospheric pressure.

It is usually considered that 10 sq ft of boiler heating surface will produce a rated boiler horsepower. A rated boiler horsepower in turn can carry a design load of from 100 to 140 sq ft of equivalent radiation. It is apparent, therefore, that 1 sq ft of boiler heating surface can carry a design load of from 10 to 14 sq ft of equivalent radiation, or somewhat more if the boiler is forced above rating. The application of these values is discussed under the heading Selection of Boilers.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel to the calorific value of 1 lb of fuel.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[7]. For general information on heating efficiencies see Chapter 29.

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated design load is the sum of the following three items[8]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

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[7]See A.S.H.V.E. research papers entitled Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931), and A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

[8]See A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.
3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932.)

The estimated maximum load is given by[8]:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 2.

TABLE 2. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS[a, b, c]

Design Load (Representing Summation of Items 1, 2, and 3,[d]		Percentage Capacity to Add for Warming Up
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggert (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired solid fuel boilers. A factor of 25 per cent over design load is adequate when oil or gas are used as fuels.

[d]240 Btu per square foot.

Other things to be considered are:

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.
6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.
7. Combustion space in the furnace.
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown, headroom in the boiler room.

Radiation Load

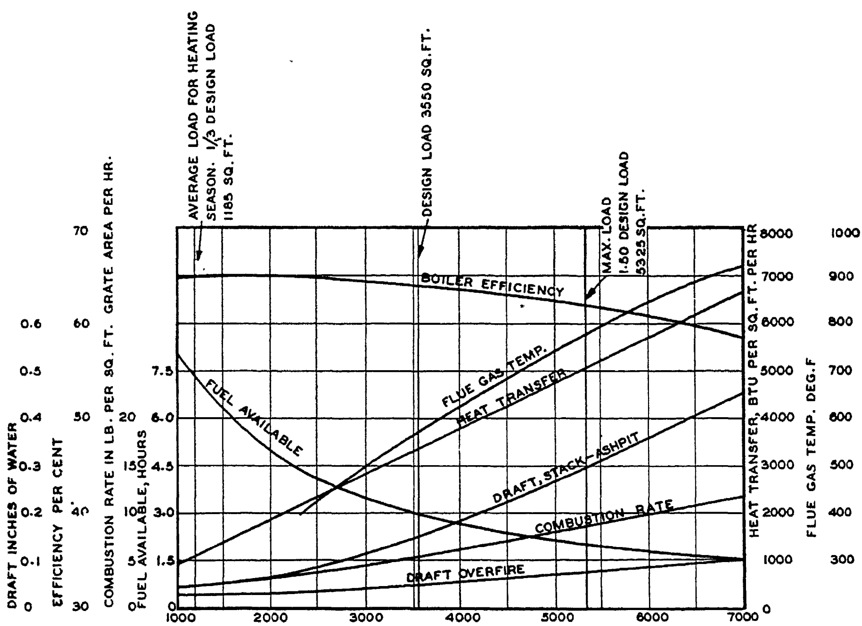
The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 5, 6 and 7, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 30. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 30.)

Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration in the various spaces of the building to be heated, which reserve, however,

is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 6.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The



BOILER DATA—

GRATE AREA, SQ. FT.	18.0
HEATING SURFACE, SQ. FT.	254
WEIGHT, LB.	9160
FUEL CAPACITY, LB.	651
FUEL AVAILABLE, LB.	414
FUEL DEPTH, IN	10

FUEL—BITUMINOUS 3/4" LUMP

ANALYSIS—	
VOLATILE MATTER	34.68%
FIXED CARBON	55.44
ASH	9.87
SULPHUR	2.86
MOISTURE	3.00
BTU PER LB.	13,655

FIG. 1. TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 35.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of 25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply

and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 36. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

Warming-up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the allowance to be made should be selected from Table 2 and should be applied to the estimated design load as determined by Items 1, 2 and 3.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 1. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 411).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of

500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500,000}{6 \times 13,000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 1. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with factors given in Table 1, Chapter 28, which include an allowance for heating up cold radiation, and for the piping tax. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is underestimated or more is added in the future.

Conversions

In the case of a solid fuel boiler converted to gas burning, the heat units supplied in the gas should be approximately twice the connected heating load. A combustion efficiency of 75 per cent for a conversion installation would provide a boiler output of $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pick-up. The presumption for a conversion job is that the boiler already is installed and probably will not be made larger; therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. The conversion of a coal or oil boiler to gas burning is accomplished much more rapidly than the reverse since but little furnace volume need be provided for the proper combustion of gas.

Other Considerations in Selection of Boilers

As it will usually be found that several boilers will meet the specifications, the final selection of the boiler may be influenced by other considerations, some of which are:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Boiler rooms should, if possible, be situated at a central point with

respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flues, and should be at least 3 ft greater than the length of the boiler firebox.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts, and with large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating, especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

CONNECTIONS AND FITTINGS

The velocity of flow through the outlets of low pressure steam heating boilers should not exceed 15 to 25 fps if fluctuation of the water line and undue entrainment of moisture are to be avoided. Steam or water outlet connections preferably should be the full size of the manufacturers' tapping and should extend vertically to the maximum height available above the boiler. For gravity circulating steam heating systems, it is recommended that a Hartford Loop, described in Chapter 32, be utilized in making the return connection.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. *I.P.S.*[9] if the gage is more than 5 ft from the boiler connection, and also in any case where the connection is less than ½ in. *I.P.S.*

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be direct connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

[9] *A.S.M.E.* Code, Identification of Piping Systems.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot water heating systems, see Chapters 32 and 33 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted up from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer always should be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

The following precautions should be taken in all installations to prevent damage to the boiler:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.

4. Condensation must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft due to air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; (*e*) improper reduction in breeching size.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a

surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb, close blow-off, draw the fire or stop burner, and open drain valve. After boiler has cooled partly, fill and flush out several times before filling it to proper water level for normal service. The use of acids, alkalis and salts for cleaning is not favored by boiler manufacturers because of the difficulty of complete removal and the possibility of subsequent injury.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

When soda ash solution is to be used the procedure is to add about 5 lb of soda ash for each 1000 sq ft of connected radiation. Fill the boiler with water until it just overflows from the surface blow outlet pipe and then fire sufficiently to raise the water temperature to the boiling point without getting up steam pressure. Crack the boiler feed valve so that a steady trickle will run out of the overflow pipe. Allow the boiler to simmer from 2 to 4 hours. At the end of this time the grease and sediment should have passed off through the overflow pipe or loosened sufficiently to drain off through the bottom blow. Extinguish the fire—preferably by letting it burn out and then dumping any live coals into the ashpit where water can be applied with a hose—and open the bottom blow wide. Rinse with fresh water and refill to the normal water level. If the water in the gage glass then does not show clear, repeat the process using a stronger soda ash solution and boiling for a longer time. It sometimes is necessary to repeat this process several times to completely rid the boiler of grease. Failure to thoroughly eliminate grease usually results in an unsteady water line and danger of damaging the boiler through having the crownsheet uncovered.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

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PROBLEMS IN PRACTICE

1 ● What is meant by a low pressure heating boiler?

A low pressure heating boiler is a boiler designed to be operated at less than 15 lb steam pressure or 30 lb water pressure as measured by a gage attached directly to the boiler.

2 ● Name the construction materials that distinguish two types of low pressure heating boilers.

a. Cast-iron.

b. Steel.

3 ● What is the normal rating range of each type of boiler?

a. Cast-iron boilers are rated at from 200 to 18,000 sq ft EDR.

b. Steel boilers are rated at from 300 to 50,000 sq ft EDR.

4 ● a. What is meant by direct boiler heating surface?
b. What is meant by indirect boiler heating surface?

a. Direct boiler heating surface is that boiler surface upon which the fire shines, namely, the walls of the firebox and the crown sheet.

b. Indirect boiler heating surface is that boiler surface not exposed to the direct rays of the fire and over which heated gases pass after they have been in contact with the direct surface. Indirect surface is generally known as convective surface.

5 ● What is the average heat transmission rate in heating boilers in Btu per sq ft of heating surface per hour?

3500 for coal burning boilers; 4200 for oil burning boilers.

6 ● What factors contribute to economical fuel operation in low pressure boilers burning coal or oil?

a. Proper furnace volume for complete combustion.

b. Arrangement of heating surfaces in series to create a turbulent and scrubbing contact of gases against the convective surfaces.

c. Rapid internal water circulation which will remove steam bubbles from the water side of heating surfaces and allow other steam bubbles to be formed. Rapid disengagement of steam bubbles increases the steam generating efficiency of each unit area of heating surface, and thereby lowers flue gas temperatures.

7 ● What equipment is usually directly attached to a low pressure heating boiler?

For coal burning steam boilers: water column, water gage, tri-cocks, steam gage, lever pop safety valve, boiler damper regulator.

For coal burning hot water boilers: damper regulator, altitude gage, thermometer, relief valve.

For oil burning boilers, the damper regulators are omitted and the following additional equipment is usually attached: automatic water feeder, low water cutout, a pressure control, and a water temperature control These are generally furnished by the oil burner manufacturer and do not come with the boiler.

8 ● What general precautions regarding the boiler should be taken to make sure a proposed heating installation will work properly?

a. Select the right size and type of boiler.

b. Be sure the combustion space is proper for the type of fuel burned.

c. Allow sufficient space around the boiler for cleaning.

d. Secure proper height and area of chimney and connecting breeching.

e. Clean the boiler thoroughly and provide surface blowoff connections and bottom blowoff connections for periodic cleaning after operation is begun.

f. See that the boiler heating surface is cleaned at regular periods.

g. Check flue gas temperatures and make a flue gas analysis at least once a month.

h. Secure information and advice from boiler manufacturer.

9 ● Below what temperature should the water in direct water heaters be maintained to reduce scale formation and corrosion?

140 F.

10 ● a. What is the heat equivalent of a boiler horsepower?
b. How many square feet of heating surface are usually required per boiler horsepower?

a. 33,471.9 Btu per hour.
b. 10 sq ft.

11 ● What is meant by equivalent evaporation?

The amount of water that a boiler would evaporate per hour, if the feed water were at 212 F and if the steam were evaporated at that temperature; this is usually spoken of as "from and at 212 F."

12 ● What loads must be considered in determining the boiler capacity required for a given installation?

Radiation load.
Hot water supply load.
Piping tax.
Warming-up allowance.
Load allowance for inefficient firing.

13 ● A boiler has 6 sq ft of grate area with a possible depth of fuel bed of 18 in. The fuel burned is bituminous coal with a heat value of 12,500 Btu per lb. The efficiency is assumed to be 50 per cent. How great a maximum load will this boiler carry if it is to be fired every 8 hours and if 20 per cent of the fuel is to be left over to kindle the next charge?

Volume of fuel bed $= 6 \times 1.5 = 9.0$ cu ft.
Available volume $= 0.80 \times 9.0 = 7.2$ cu ft.
Weight of available fuel $= 40 \times 7.2 = 288.0$ lb.

Fuel burned per hour $= \frac{288.0}{8} = 36.0$ lb.

Heat released $= 36.0 \times 12{,}500 \times 0.50 = 225{,}000$ Btu per hour.

Maximum load $= \frac{225{,}000}{240} = 938$ equivalent square feet.

14 ● What are the usual causes of unsteady water line and priming?

Grease and dirt in boiler.
Overload, resulting in insufficient steam liberating area.
Small outlet connections.

15 ● What type of return connection can be used for gravity steam heating systems to make the use of check valves unnecessary?

The Hartford Loop. (See Chapter 32.)

Chapter 26

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Chimney Construction, Chimneys for Gas Heating

THE design and construction of a chimney is so important a part of the heating engineer's work that a general knowledge of draft characterictics and calculations is essential.

Draft, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, *viz*, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all of the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above their normal rating. Natural draft systems have been, and are still being, employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of

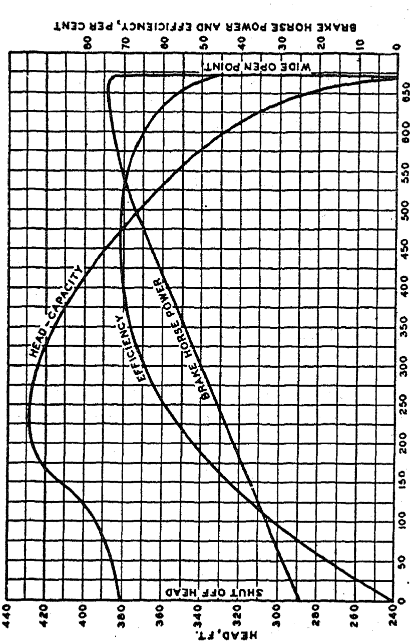


FIG. 2. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

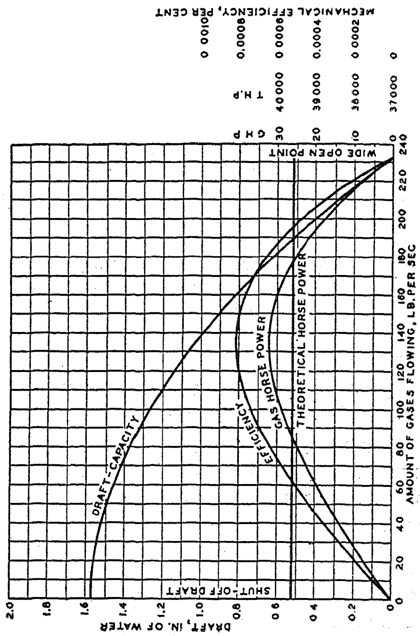


FIG. 3. OPERATING CHARACTERISTICS OF TYPICAL NATURAL DRAFT CHIMNEY

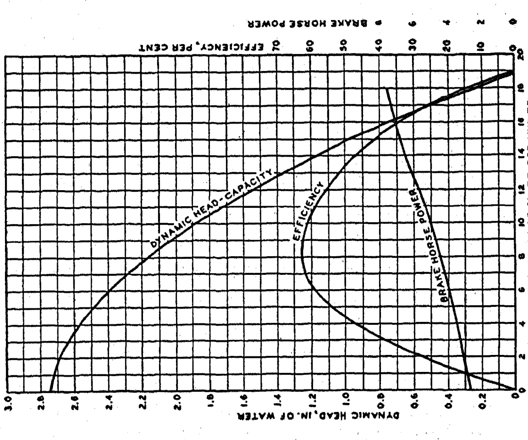


FIG. 1. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

operation is increased above their normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. Mechanical draft includes the induced and Venturi types of draft systems in which the pressure difference is the result of a suction and also the forced draft system in which the pressure difference is the result of a blowing. Mechanical draft systems tend to produce a vacuum or a plenum, according as the system used in its production creates a pressure difference below, or above, atmospheric pressure, respectively. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it is advantageous to compare its general operating characteristics with those of a centrifugal pump and also a centrifugally-induced draft fan, there being a close similarity among the three. Figs. 1, 2 and 3 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to the head-capacity curve of the pump and also to the dynamic-head capacity of the fan; the efficiency curve of the chimney to the efficiency curves of the pump and fan; and the gas horsepower curve of the chimney to the brake horsepower curves of the pump and fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft*. When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the *available draft*. The general equation for the available draft intensity of a natural draft chimney with a circular section is as follows:

$$D_a = 2.96HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{D^5B_oW_c} \tag{1}$$

where

D_a = available draft, inches of water.
H = height of chimney above grate bars, feet.
B_o = barometric pressure corresponding to altitude, inches of mercury.
W_o = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.
W_c = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
T_o = absolute temperature of atmosphere, degrees Fahrenheit.
T_c = absolute temperature of chimney gases, degrees Fahrenheit.
W = amount of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.
f = coefficient of friction.
L = length of friction duct of the chimney, feet.
D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_o = 29.92 in. of mercury; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft. The chimney discharges 100 lb of gases per second.

Substituting these values in Equation 1 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 4 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second, at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 4.

The efficiency of a natural draft chimney is the thermodynamical ratio of the energy output to the energy input. The energy output is the total

work done by the chimney in moving the gases and corresponds to the water horsepower of a centrifugal pump, or the total work done by a fan in moving the gases. The energy input is equal to the theoretical amount of power generated by the chimney and corresponds to the power input of the driving unit of a centrifugal pump or a fan. The thermodynamical efficiency is given by the equation:

$$E_t = \frac{K_a W D_a}{A\sqrt{H}} \tag{2}$$

where

K_a = a constant depending upon the temperature of the gases, the atmospheric temperature, the elevation of the plant, and the density and specific heat of the gases. For average operating conditions, K_a = 0.0065.

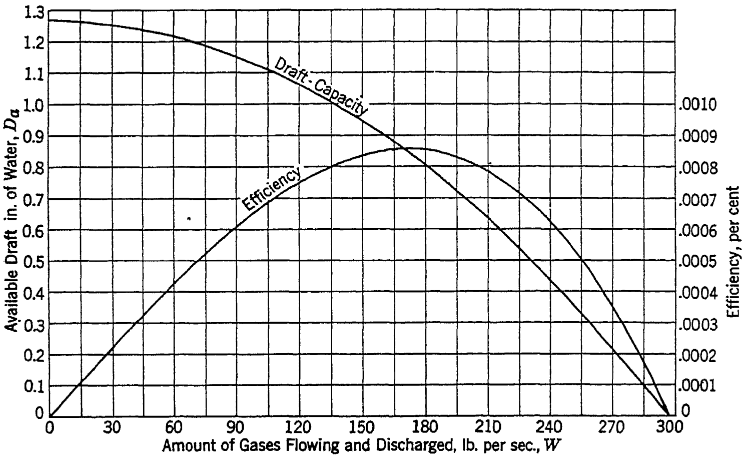


FIG. 4. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

Fig. 4 shows the variation in the efficiency of the chimney under consideration for the operating conditions noted. This curve rises from zero at shut-off draft to a maximum for a certain draft and its corresponding capacity and then drops again to zero at the wide open point. The point of maximum efficiency is located by the point on the draft-capacity curve equal to two-thirds of the theoretical draft intensity. In Example 1 the maximum efficiency is at an available draft intensity of $\frac{2}{3} \times 1.27 = 0.85$ in. of water and the corresponding capacity of 175 lb per second.

The efficiency curve of a natural draft chimney corresponds to the efficiency curves of a centrifugal pump and a fan and serves the same general use in that it locates the region of most economical operation. In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure* varies inversely as the altitude of the plant above sea level. Fig. 5 gives the barometric pressure corresponding to various elevations as computed from the equation:

$$E_1 = 62{,}737 \log \frac{29.92}{B_o} \tag{3}$$

where

E_1 = altitude of plant above sea level, feet.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 F and sea level barometric pressure is given by the equation:

$$W_C = 0.131 CO_2 + 0.095\ O_2 + 0.083\ N_2 \tag{4}$$

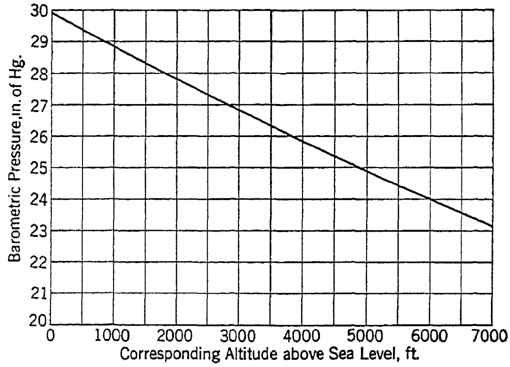


FIG. 5. RELATION BETWEEN BAROMETRIC PRESSURE AND ALTITUDE

In this equation CO_2, O_2 and N_2 represent the percentages of the parts by weight of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_C may be assumed at 0.09.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* does not vary appreciably from the gas temperature as it leaves the breeching and enters the chimney. For average operating conditions, the chimney gas temperature will vary between 500 F and 650 F except in the case when economizers and recuperators are used, when the temperature will vary between 300 F and 450 F. If a chimney has been properly constructed, properly lined and has no air infiltration due to open joints, the temperature of the gases throughout the chimney will not differ appreciably from the foregoing figures. In most up-to-date heating plants, the temperature may be read from instruments or ascertained from a pyrometer.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been found to be approximately 0.016. This factor, of course, will be much less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the material of which they are constructed becomes covered with a layer of soot and the coefficient of friction should be the same for all types of chimneys.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. The *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the boiler. The total products of combustion may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \tag{5}$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.
G = total grate surface of boilers, square feet.
W_{tp} = total weight of products of combustion per pound of fuel.

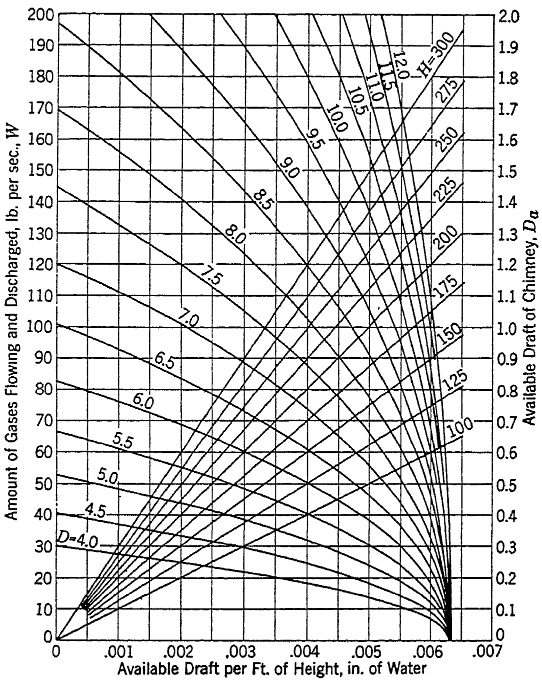


FIG. 6. CHIMNEY PERFORMANCE CHART

Fig. 6 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific operating conditions, a new chart should be constructed from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of

chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft chimney are given by the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{6}$$

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{7}$$

where

H = required height of chimney above grate bar level, feet.
D = required minimum diameter of chimney, feet.
V = chimney gas velocity, feet per second.
D_r = total required draft demanded by the entire installation outside of the chimney, inches of water.

Equations 6 and 7 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimneys as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will cost least. Since the cost of a chimney structure, regardless of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi tHD \tag{8}$$

where

Q = volume of material, cubic feet.
t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor

HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of *HD* for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product *HD* will be least. The solution is obtained by equating the

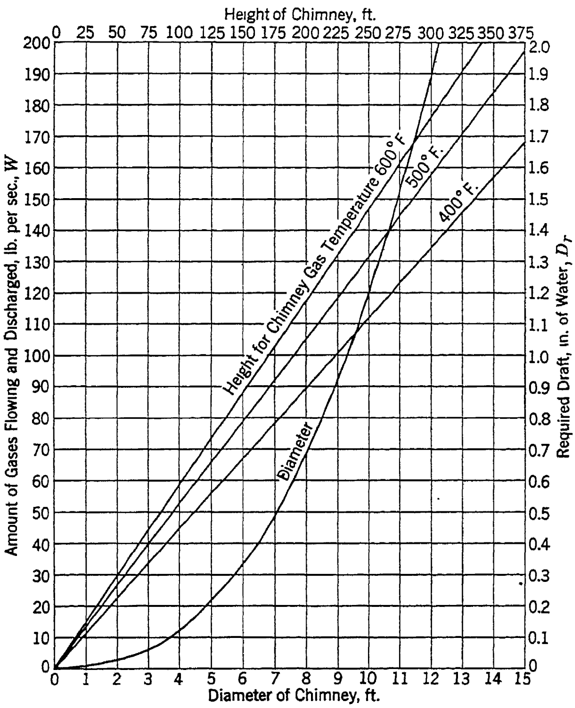


FIG. 7. ECONOMICAL CHIMNEY SIZES

product of Equations 3 and 4 to *HD*, differentiating this product with respect to *V* and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772T_c\left(\dfrac{W_o}{T_o} - \dfrac{W_c}{T_c}\right)\sqrt{\dfrac{WT_c}{B_oW_c}}}{fW_c} \right)^{2/5} \qquad (9)$$

where V_e = economical chimney gas velocity, feet per second.

Equation 9 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the

corresponding height and diameter can then be determined from Equations 6 and 7, respectively, and the economical size will then be attained. Equations 6, 7 and 9 may be simplified considerably for average operating conditions in an average size steam plant by assuming the following conditions:

Average chimney gas temperature, 500 F	$T_c = 960$
Mean atmospheric temperature, 62 F	$T_o = 522$
Average coefficient of friction, 0.016	$f = 0.016$
Average chimney gas density, 0.09	$W_c = 0.09$
Sea level elevation with barometer of 29.92	$B_o = 29.92$

Substituting these values in Equations 9, 7 and 6, respectively, and reducing:

$$V_e = 13.7W^{1/5} \tag{10}$$

$$D = 1.5W^{2/5} \tag{11}$$

$$H = 190D_r \tag{12}$$

Fig. 7 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 10, 11 and 12. They are based on the operating factors used in reducing Equations 6, 7 and 9 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 9. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \tag{13}$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water.
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators, or air heaters, inches of water.

The left hand member of Equation 13 represents the total amount of available draft created by the pressure transformer, that is, the natural

TABLE 1. RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES[a]

Warm Air Furnace Capacity in Sq In. of Leader Pipe	Steam Boiler Capacity Sq Ft of Radiation	Hot Water Heater Capacity Sq Ft of Radiation	Nominal Dimensions of Fire Clay Lining in Inches	Rectangular Flue: Actual Inside Dimensions of Fire Clay Lining in Inches	Rectangular Flue: Actual Area Sq In.	Round Flue: Inside Diameter of Lining in Inches	Round Flue: Actual Area Sq In.	Height in Ft Above Grate
790	590	973	8½ x 13	7 x 11½	81			35
1000	690	1,140				10	79	
	900	1,490	13 x 13	11¼ x 11¼	127			
	900	1,490	8½ x 18	6¾ x 16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13 x 18	11¼ x 16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18 x 18	15¾ x 15¾	248			
	2,480	4,090	20 x 20	17¼ x 17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20 x 24	17 x 21	357			
	5,000	8,250	24 x 24	21 x 21	441			55
	5,570	9,190		24 x 24[b]	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24 x 28[b]	672			
	8,700	14,400		28 x 28[b]	784			
	9,380	15,500				27	573	
	10,150	16,750		30 x 30[b]	900			
	10,470	17,250		28 x 32[b]	896			

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

[b]Dimensions are for unlined rectangular flues.

draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft intensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \qquad (14)$$

where

D_a = available draft intensity, inches of water.
D_r = required draft, inches of water.

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed,

coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 8 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals,

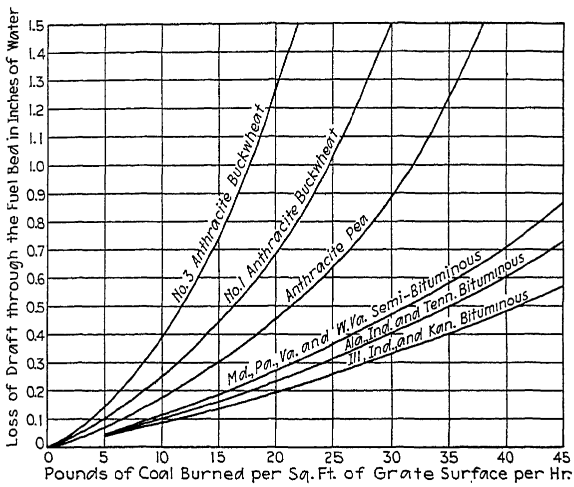


Fig. 8. Draft Required at Different Rates of Combustion for Various Kinds of Coal

much less draft is required than when the particles are small and are well broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss

through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_B) also varies between wide limits and, in general, depends upon the following factors:

1. Type of boiler.
2. Size of boiler.
3. Rate of operation.
4. Arrangement of tubes.
5. Arrangement of baffles.
6. Type of grate.
7. Design of brickwork setting.
8. Excess air admitted.
9. Location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. The amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) is given by the general equation:

$$h_{Br} = \frac{0.000194 W^2 T_C f L}{A^2 B_O W_C C_{br}} \tag{15}$$

where

W = the amount of gases flowing, pounds per second.
T_c = absolute temperature of breeching gases, degrees Fahrenheit.
f = coefficient of friction.
L = length of breeching, feet.
A = area of breeching, square feet.
B_o = atmospheric pressure corresponding to altitude, inches of mercury.
W_c = weight of a cubic foot of breeching gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
C_{br} = hydraulic radius of breeching section.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194 W^2 T_c}{A^2 B_o W_c} \tag{16}$$

and represents the amount of draft required to accelerate the gases from zero velocity to the velocity at which the gases are flowing, or in other words, from a static gas condition of zero flow to the amount of gases flowing throughout the installation. This loss corresponds to the velocity head in water works systems.

The *draft loss due to bends* (h_{Bd}) is equivalent to the loss due to the velocity head for a 90-deg bend. In changing direction of flow, the gas velocity decreases to zero with a loss of velocity head and then increases to its proper value at the expense of a loss in pressure head, the net result being a loss in pressure head equal to the velocity head at the bend. This loss is given by the equation:

$$h_{Bd} = \frac{0.000194 W^2 T_c}{A^2 B_o W_c} \tag{17}$$

The friction at a right-angle bend is sometimes expressed as the equivalent of a straight length of flue of a certain length for a certain diameter, similar to the procedure used in estimating the loss due to bends in piping systems conducting water. Most flues, however, particularly breechings, are built square or rectangular in section and no general equation based on the shape of the flue can be conveniently expressed.

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194 K_C W^2 T_C}{A_s^2 B_O W_C} \qquad (18)$$

where

K_C = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section.

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_O W^2 T_C}{A_s^2 B_O W_C} \qquad (19)$$

where

K_O = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section.

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_C}{10^{12}} \qquad (20)$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large

plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters*. Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The construction, location, height and area of the chimney to which a heating boiler or warm-air furnace is connected affect the operation of the entire heating system. Most residence chimneys are built of brick and may be either lined or unlined, but in either case the *walls must be air-tight* and there should be *only one smoke opening* into the chimney. Cleanout, if provided, must be *absolutely air-tight* when closed.

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining. Fire-clay flue linings shall be manufactured from suitable refractory clay, either natural or compounded, and shall be adapted to withstand high temperatures and the action of flue gases. They shall be of standard commercial thickness, but not less than ¾ in. All fire-clay flue linings shall meet the standard specification of the *Eastern Clay Products Association*. The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining. The wash or splay shall be formed of a rich cement mortar. To improve the draft the wash surface should be concave wherever practical.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped with stone, terra cotta, concrete, cast-iron, or other approved material; but no such cap or coping shall decrease the flue area.

There shall be but one connection to the flue to which the boiler or furnace smoke-pipe is attached. The boiler or furnace smoke-pipe shall be thoroughly grouted into the chimney and shall not project beyond the inner surface of the flue lining.

The size or area of flue lining or of brick flue for warm-air furnaces depends on height of chimney and capacity of heating system. For chimneys not less than 35 ft in height above grate line, the net internal dimensions of lining should be at least 7 x 11½ in. for a total leader pipe area up to 790 sq in. Above 790 and up to 1,000 sq in. of leader pipe area the lining should be at least 11¼ x 11¼ in. inside. In case of brick flues not less than 35 ft in height with no linings, the internal dimensions should be at least 8 x 12 in. up to 790 sq in. of leader area, and at least 12 x 12 in. for leader capacities up to 1,000 sq in. Chimneys under 35 ft in height are unsatisfactory in operation and hence should be avoided.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of

burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

A study of a typical *back-draft diverter* (Fig. 9) shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely

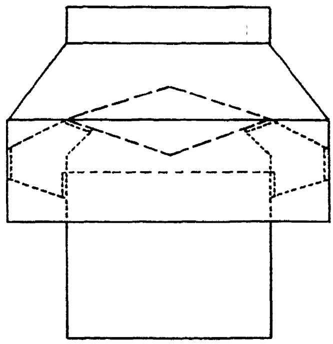


FIG. 9. TYPICAL BACK-DRAFT DIVERTER

under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value. Converted boilers or furnaces, as well as gas-designed appliances, should be provided with back-draft diverters.

As is the case with the complete combustion of almost all fuels, the products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air into the chimney through the openings in the back-draft diverter, lowers the dew point of the mixture, and reduces the tendency of the water vapor to condense.

A chimney for a gas-fired boiler or furnace should be constructed in accordance with the principles applicable to other boilers. Where the wall forming a smoke flue is made up of less than an 8-in. thickness of brick, concrete, or stone, a burnt fire clay flue tile lining should be used. Care should be used that the lengths of flue tile meet properly with no openings at the joints. Cement mortar should be used for the entire chimney.

TABLE 2. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

Height of Chimney Feet	Gas Consumption in Thousands of Btu per Hour								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8.00	9.40	10.50	12.35	13.85
40	4.25	5.50	6.40	7.10	7.80	9.15	10.25	12.10	13.55
60	4.10	5.35	6.20	6.90	7.60	8.90	10.00	11.85	13.25
80	4.00	5.20	6.00	6.70	7.35	8.65	9.75	11.50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

Table 2 gives the minimum cross-sectional diameters of round chimneys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

The flue connections from a gas-fired boiler or furnace to the chimney should be of a non-corrosive material. In localities where the price of gas requires the use of highly efficient appliances, the material used for the flue connection not only should be resistant to the corrosion of water, but should resist the corrosion of dilute solutions of sulphur trioxide in water. Sheet aluminum, as well as some other materials, seems to serve this purpose very well.

PROBLEMS IN PRACTICE

1 ● What is draft?

Draft is an unbalanced pressure between the atmosphere and the passages in the apparatus or construction through which the gases flow.

2 ● What two kinds of draft need be considered?

Natural draft caused by temperature differences, and artificial draft caused by mechanical forcing.

3 ● What is the effective height of a chimney?

The height from the grate level to the top of the chimney is the effective height in producing natural draft.

4 ● What dual purpose does a tall chimney fulfill?

A tall chimney primarily creates the necessary draft to move the air required for the combustion process and to move the products of combustion, and secondarily it discharges the gases at a high elevation to prevent them from becoming a nuisance.

5 ● a. Name the principal advantages of natural draft.
b. Name the principal disadvantages of natural draft.

a. Simplicity, reliability, freedom from mechanical parts, low cost of maintenance, relatively long life, relatively low depreciation, operation with no power requirement.

b. Lack of flexibility, irregularity, dependence on surroundings, susceptibility to temperature changes.

6 ● How is mechanical draft created?

By forced draft, by induced-draft fans, or by a Venturi chimney.

7 ● Distinguish between theoretical and available draft.

Theoretical draft is the difference in pressure inside and outside the base of a chimney when it is under operating temperatures but when there are no gases flowing. Available draft is less than theoretical draft by the friction loss due to the flow of gases through the chimney.

8 ● Explain the term efficiency of a natural draft chimney.

The efficiency of a chimney is the ratio of the work it does in moving gases to the theoretical amount of power it generates.

9 ● How is the available draft used in a heating plant?

The available draft at the base of the chimney is used to overcome the loss in pressure through the grate, the fuel bed, the boiler passes, the breeching, and the chimney.

10 ● What are some of the factors that influence the draft loss through the fuel bed?

Uniformity and size of coal, the amount of ash mixed with the fuel on the grate, thickness of fuel bed, rate of combustion, amount of air supply as related to the coal burning rate.

11 ● How does the volatile matter content affect the draft loss through the fuel bed?

The higher the volatile content and the lower the fixed carbon content, the lower the draft loss.

12 ● In what cases will there be no fuel bed draft loss?

In oil, gas, and powdered fuel firing the fuel is mixed and burned in suspension; consequently, no measurable resistance is encountered in the combustion zone.

13 ● Is it possible to state an average value for the draft loss through a boiler and its setting?

No. The draft loss varies widely and depends on many factors such as the size and type of gas passageways. The manufacturer is usually able to supply such information.

14 ● Of what significance is the CO_2 content of stack gases in establishing draft loss?

The CO_2 content of the exit gases is a measure of the completeness of the combustion and the amount of excess air supplied. Low CO_2 indicates a high excess of air and hence a high draft loss.

15 ● What two effects does an economizer have on the draft loss?

An economizer offers resistance to the flow of gases over the added surfaces; it lowers the temperature of the gases going to the chimney and therefore decreases the available draft. This decrease often necessitates the addition of forced draft.

16 ● What main provisions should be considered in good chimney construction?

Chimneys should be air-tight and connected to only one smoke opening. The chimney top should be high enough above surroundings so the wind will not strike it at any angle

above the horizontal. Chimney walls should be not less than one brick in width, and they should be lined with fire clay tile of the size required for the attached heating unit. Tile lining sizes are stated as outside dimensions; therefore, their effective dimensions are less by the thickness of the wall.

17 ● What is the purpose of a back draft diverter as used on gas burning units?

Since the fuel is supplied under pressure independent of draft it is necessary to free the unit from the variable chimney draft and to supply air for combustion in direct proportion to the supply of fuel gas. The back draft diverter protects the pilot and burners from down drafts.

Chapter 27

FUELS AND COMBUSTION

Classification of Coal, Air for Combustion, Draft Required, Combustion of Anthracite, Firing Bituminous Coal, Burning Coke, Hand Firing, Classification and Use of Oil, Classification and Use of Gas

THE choice of fuel for heating is a question of economy, cleanliness, fuel availability, operation requirements, and control. The principal fuels to be considered are coal, oil, and gas.

COAL

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in *U. S. Bureau of Mines Bulletin* 276.

A classification of coals is given in Table 1, and a brief description of the kinds of fuels is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other:

Anthracite is a clean, dense, hard coal which creates very little dust in handling. It is comparatively hard to ignite but it burns freely when well started. It is non-caking, it burns uniformly and smokelessly with a short flame, and it requires little attention to the fuel beds between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces.

Semi-anthracite has a higher volatile content than anthracite, it is not as hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling it. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal*.

The term *bituminous coal* covers a large range of coals and includes many types having distinctly different composition, properties, and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite

easily and burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible especially at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and it also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ash pit continue to burn. Very little smoke or soot is formed.

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

TABLE 1. CLASSIFICATION OF COALS BY RANK[f]

Legend: F.C. = Fixed Carbon. V.M. = Volatile Matter. Btu = British thermal units.

CLASS	GROUP	LIMITS OF FIXED CARBON OR BTU MINERAL-MATTER-FREE BASIS	REQUISITE PHYSICAL PROPERTIES
I. Anthracite	1. Meta-anthracite	Dry F.C., 98 per cent or more (Dry V.M., 2 per cent or less)	
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	Non-agglutinating[a]
II. Bituminous[c]	1. Low volatile bituminous coal	Dry F.C., 77 per cent or more and less than 86 per cent (Dry V.M., 23 per cent or less and more than 14 per cent)	
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 77 per cent (Dry V.M., 31 per cent or less and more than 23 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); and moist[b] Btu, 14,000[d] or more	
	4. High volatile *B* bituminous coal	Moist[b] Btu, 13,000 or more and less than 14,000[d]	
	5. High volatile *C* bituminous coal	Moist Btu, 11,000 or more and less than 13,000[d]	Either agglutinating or non-weathering[e]
III. Sub-bituminous	1. Sub-bituminous *A* coal	Moist Btu, 11,000 or more and less than 13,000[d]	Both weathering and non-agglutinating
	2. Sub-bituminous *B* coal	Moist Btu 9500 or more and less than 11,000[d]	
	3. Sub-bituminous *C* coal	Moist Btu 8300 or more and less than 9500[d]	
IV. Lignitic	1. Lignite	Moist Btu less than 8300	Consolidated
	2. Brown coal	Moist Btu less than 8300	Unconsolidated

[a]If agglutinating, classify in low-volatile group of the bituminous class.

[b]Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[c]Pending the report of the Subcommittee on Origin and Composition and Methods of Analysis, it is recognized that there may be non-caking varieties in each group of the bituminous class.

[d]Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[e]There are three varieties of coal in the High-volatile *C* bituminous coal group, namely, Variety 1, agglutinating and non-weathering; Variety 2, agglutinating and weathering; Variety 3, non-agglutinating and non-weathering.

[f]Adapted from A.S.T.M. Standards on Coal and Coke, p. 68, *American Society for Testing Materials*, Philadelphia, 1934.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke*. The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

The sale of *petroleum cokes* for domestic furnaces has been small and is generally confined to the Middle West. They vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

In order to obtain perfect combustion a definite amount of air is required for each pound of fuel fired. A deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack.

Total Air Required. The theoretical amount of air required per pound of fuel for perfect combustion is dependent upon the analysis of the fuel;

TABLE 2. POUNDS OF AIR PER POUND OF FUEL AS FIRED

ANTHRACITE	COKE	SEMI-BITUMINOUS	BITUMINOUS	LIGNITE
9.6	11.2	11.2	10.3	6.2

however, for estimating purposes the theoretical air required for different grades of fuel may roughly be taken from Table 2. An excess of about 50 per cent over the theoretical amount is considered good practice under usual operating conditions.

The amount of excess air, based upon the laws of combustion, can be determined by its relation to the percentage of CO_2 (carbon dioxide) in the products of combustion. This relationship is shown by the curves (Fig. 1) for high and low volatile coals and for coke. In hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

Secondary Air. The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed, and diameter of fire pot. The ratio of the secondary to the primary air increases with decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the fire pot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not

so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel bed, which draws in more secondary air through the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the

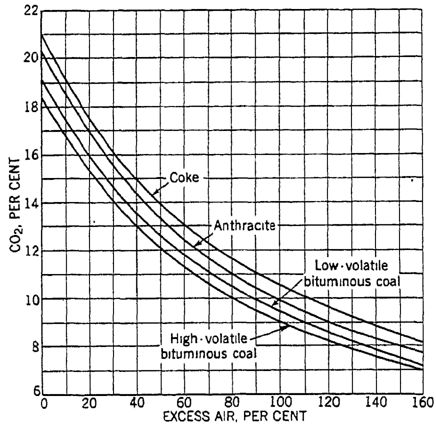


FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

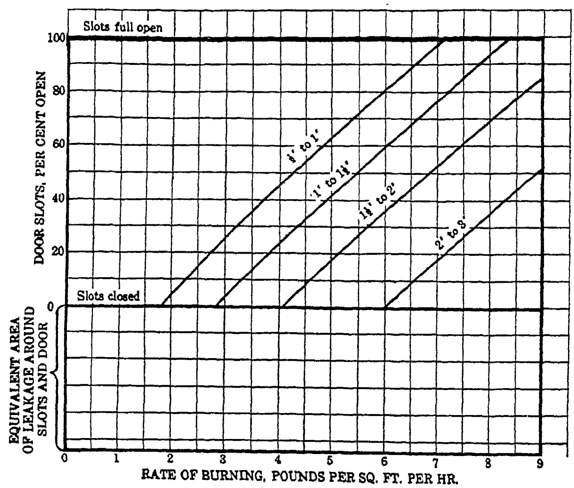
size of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

Fig. 2, taken from the *U. S. Bureau of Mines Report of Investigations* No. 2980, shows the relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning; these openings are with the ashpit damper wide open, and would be less if the available draft permits of its being partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial. The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.

2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.



From U. S. Bureau of Mines.

FIG. 2. RELATIVE AMOUNT OF FIRE DOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.

4. For ordinary house operation, secondary air is needed after each firing for about one hour.

Draft Required

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning, and it is often possible to produce a higher rate by increasing the thickness of the fuel bed.

Combustion of Anthracite[1]

An anthracite fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel

[1]See reports published by *The Anthracite Institute Laboratory*, Primos, Pennsylvania.

can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and for best results this coal should be fired deeply.

Stove size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 12 in. deep. The only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for firepots up to 20 in. in diameter, with a depth of from 10 to 15 in.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open, the cold-air check closed, and by controlling the fire with the air-inlet damper only. Pea size can also be fired in layers with stove or egg size anthracite and its use in this manner will reduce the fuel costs and attention required.

Buckwheat size coal requires much the same attention as pea size coal, except that the smaller size of the fuel makes it more difficult to burn on ordinary grates. Even greater care must be taken in shaking the grates than with pea coal on account of the danger of the fuel falling through the grate. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. Where frequent attention can be given and where there is not a big heat demand, this fuel is frequently burned without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and special grates or retorts frequently are used with this fuel for best results.

No. 2 buckwheat anthracite, or rice size, is used only with forced draft equipment on mechanical stokers. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Bituminous Coal

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire door or by opening the fire door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the firebox, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the firebox and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the firebox so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The fuel bed should be carried as deep as the size of fuel and the available draft permit, in order to have as much coked fuel as possible for pushing to the rear of the firebox at the time of firing. A deep fuel bed allows the longest firing intervals.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since soft coal burns more rapidly than hard coal and with less draft. Soft coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

Semi-bituminous coal is fired as bituminous coal, and because of its caking characteristics it requires practically the same attention. The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recom-

mended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

Burning Coke

Coke is a very desirable fuel and usually will give satisfaction as soon as the user learns how to control the fire. Coke ignites and burns very rapidly with less draft than anthracite coal. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds more rapidly than an anthracite fire to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. A deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should be shaken only until the first red particles drop from the grates in cold weather. Since coke weighs only about half as much as anthracite per cubic foot only about half as much can be put in the firepot, so it will be necessary to fire oftener. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large firepots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a coke of uniform size is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or broken up before using.

Dustless Coal

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with a solution of calcium chloride or a mixture of calcium and magnesium chlorides. Both these salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery and in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is sometimes treated at the mine, but more usually by the local distributor just before delivery. The solution is sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size.

Pulverized Coal

Installations of pulverized coal burning plants in heating boilers are of the unit type, in which the pulverized coal is delivered into the furnace immediately after grinding, together with the proper amount of preheated air. With this apparatus, where the necessary furnace volume is obtainable, high efficiencies can be obtained.

A 150-hp boiler has generally been considered the smallest size for which pulverized fuel is feasible. Complications are introduced if an installation with a single boiler has to take care of very light loads.

Hand Firing

Hand firing is the oldest and the most widely used method of burning coal for heating purposes. To keep the fuel bed in proper condition where hand firing is used, the following general rules should be observed:

1. Remove ash from fuel bed by shaking the grates whenever fresh fuel is fired. This removes ash from the fire, enables the air to reach the fuel, and does away with the formation of clinker which is melted ash.

2. Supply the boiler with a deep bed of fuel. Nothing is gained by attempting to fire a small amount of fuel. A deep bed of fuel secures the most economical results.

3. Remove ash from ashpit at least once daily. Never allow ash to accumulate up to the grates. If the ash prevents the air from passing through, the grate bars will burn out and much clinker trouble will be experienced.

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly.

OIL

Uniform oil specifications were prepared in 1929 by the *American Oil Burner Association*, in coöperation with the *American Petroleum Institute*, the U. S. Bureau of Standards, the *American Society for Testing Materials* and other interested organizations. Oil fuels were classified into six groups, as indicated by Table 3. When these specifications were prepared, it was generally accepted that the first three grades were adapted to domestic use, while the last three were suitable only for commercial and industrial burners. Today domestic installations are using No. 4 of the so-called *heavy-oil* group, due principally to the fact that No. 4 oils in general are being offered of better grade and adaptability than those called for in the commercial specifications.

Since the specifications as originally drawn provide for maximum limits only for the several grades, this differentiation has not proved stable. Realizing how unsatisfactory it is to have specifications which permit the substitution of one grade for another, the U. S. Bureau of Standards in coöperation with the *American Society for Testing Materials* is figuring on a new set of specifications providing for definite limits for each grade. When these specifications are adopted, it is expected that the *National Board of Fire Underwriters* will retest all burners using oils of the maximum specifications for the grade so that if a burner is approved for a certain grade it will burn any oil meeting the specifications for that particular grade.

Several burners adapted to industrial use have recently been listed for automatic operation with No. 5 oil. Usually oils No. 5 or 6 require preheating for proper operation, but where conditions are favorable, No. 5 can be used without the equipment that this entails.

There are two reasons for the trend to lower grades of oil. While the lighter oils contain slightly more heat units per pound, the weight per

TABLE 3. COMMERCIAL STANDARD FUEL OIL SPECIFICATIONS[a]

A. Detailed Requirements for Domestic Fuel Oils

GRADE OF OIL	APPROX. BTU PER GAL.[b]	FLASH POINT Min.	FLASH POINT Max.	WATER AND SEDIMENT, MAXIMUM	POUR POINT,[c] MAXIMUM	DISTILLATION TEST		VISCOSITY MAXIMUM
No. 1 Domestic Fuel Oil A light distillate oil for use in burners requiring a high grade fuel.	139,000	110 F or legal	165 F	0.05%	15 F	10% point, maximum 420 F	End point, maximum 600 F	
No. 2 Domestic Fuel Oil A medium distillate oil for use in burners requiring a high grade fuel.	141,000	125 F or legal	190 F	0.05%	15 F	10% point, maximum 440 F	90% point, maximum 620 F	
No. 3 Domestic Fuel Oil A distillate fuel oil for use in burners where a low viscosity oil is required.	143,400	150 F or legal	200 F	0.1%	15 F	10% point, maximum 460 F	90% point, maximum 675 F	Saybolt Universal at 100 F 55 seconds

B. Detailed Requirements for Industrial Fuel Oils

GRADE OF OIL	APPROX. BTU PER GAL.[b]	FLASH POINT, MIN. MAX.	WATER AND SEDIMENT, MAXIMUM	POUR POINT,[e] MAXIMUM	VISCOSITY, MAXIMUM
No. 4. Industrial Fuel Oil An oil known to the trade as a light fuel oil for use in burners where a low viscosity industrial fuel oil is required.	144,500	150 F. See Note[d]	1.0%	See Note[e]	Saybolt Universal at 100 F 125 seconds
No. 5 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "B" for burners adapted to the use of industrial fuel oil of medium viscosity.	146,000	150 F	1.0%		Saybolt Furol at 122 F 100 seconds
No. 6 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "C" for burners adapted to oil of high viscosity.	150,000	150 F	Water sediment 1.75% 0.25%		Saybolt Furol at 122 F 300 seconds

[a]Adapted from "Fuel Oils," p. 2, *U. S. Department of Commerce, Bureau of Standards, Commercial Standard CS12-33*, Washington, 1933.

[b]Government specifications do not give Btu per gallon, but they are noted here for information only.

[c]Lower or higher pour points may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 0 F under any conditions.

[d]Whenever required, as for example in burners with automatic ignition, a maximum flash point may be specified. However, these specifications shall not require a flash point less than 250 F under any conditions.

[e]Pour point may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 15 F under any conditions.

gallon increases more rapidly than the decrease in heat units per pound, and oil is bought by the gallon. As a consequence, while a No. 1 oil may contain 139,000 Btu per gallon, oil No. 5 may test 146,000 Btu per gallon, or 6 per cent more. Usually there is a differential of 3¢ to 4¢ between the No. 1 and No. 5 oils, so that the economy of buying the heavier fuels is apparent; there remains the economic utilization of the heat content of the heavier oils.

The cost of oil fuel is dependent also upon the amount that can be delivered at one time, and the method of delivery. Common practice has split the tank of the truck delivering oils for domestic use into compartments of 150 to 500-gal capacity, and these *unit dumps* are made the basis of price. Where a truck can be connected to a storage-tank *fill* and quickly discharge its oil by pump, the price obviously can be less than where a smaller quantity must be drawn off in 5-gal cans and poured. For similar reasons an installation that can be supplied from a tank car on a siding provides for a lower unit fuel cost than one where the oil must be trucked, even in the large trucks holding 2,000 gal or more that are used for distributing the heavier oils.

GAS

Gas is broadly classified as being either *natural* or *manufactured.* Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes, and is often designated as *city gas.*

When gas is burned a large amount of water vapor is produced as one of the products of combustion. This ordinarily escapes up the chimney, carrying away with it a certain amount of heat. However, when the heat value of gas is determined in an ordinary calorimeter, this water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the gas. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heat value of gas is specified. The heat that is reclaimed by the condensation of the water vapor amounts to about 10 per cent of the total heat value. It is impractical to utilize the entire higher heat value of the gas in any house-heating appliance, because to do so it would be necessary to cool the products of combustion down below their dew point, which is ordinarily in the neighborhood of 130 F.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 700 to 1,500 Btu per cubic foot, the majority of natural gases averaging about 1,000 Btu per cubic foot. Table 4 shows typical values for the three main oil fields, although values from any one field vary materially.

Table 4 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross

or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. There are limits to the variation allowable, because the specific gravity of the gas depends on its

TABLE 4. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS, BASED ON GAS AT 60 F AND 30 IN. HG.

Gas	Btu per Cu Ft		Specific Gravity, Air = 1.00	Air Required for Combustion, (Cu Ft)	Products of Combustion				Theoretical Flame Temperature, (Deg Fahr)
					Cubic Feet			Ultimate CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—Mid-Continental	967	873	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1232	1120	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carburetted water gas	536	496	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	134	124	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

composition, and too great a change in the specific gravity necessitates a change in the adjustment of the burners of small appliances.

Table 4 shows that a large proportion of the products of combustion when gas is burned may consist of water vapor, and that the greater the proportion of water vapor, the lower the maximum attainable CO_2 by gas analysis. The table also shows that a low calorific value does not necessarily mean a low flame temperature since, for example, natural gas has a theoretical flame temperature of 3600 F and blue water gas of 3800 F, although it has a calorific value less than one third that of natural gas.

The quantity of air given in Table 4 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

PROBLEMS IN PRACTICE

1 ● Name several important properties of coal from a utilization standpoint.

a. Caking tendency, whether none, weak, or strong.
b. Quantity of volatile matter.
c. Friability.
d. Fusibility of the ash.

2 ● What are the main data commonly available that fix the qualities of coal, and do these tell the whole story?

a. Calorific value, Btu per pound.
b. Proximate analysis giving percentages of moisture, volatile matter, fixed carbon, ash, and sulphur.
c. Temperature at which the ash softens.
d. Screen sizes.

Other important qualities not usually given are the friability of the coal, its caking tendency, and the qualities of the volatile matter. The percentage of ash and its fusion temperature do not tell how the ash is distributed or how much of it is less fusible lumps of slate or shale.

3 ● Are there available complete and sufficient data on gas and oils to fix their burning properties and furnace requirements?

Yes. Because gas and oils are of simple and uniform composition, data are available to fix their burning properties and furnace requirements, but the ability to control their combustion is somewhat less determinable.

4 ● What effect does moisture in fuels have on their efficiency?

With any solid fuel, latent and sensible heat are lost at the stack when moisture is dried out of the fuel in burning, and when its hydrogen is burned. Therefore, such fuels as sub-bituminous coal and lignite, which are high in moisture content, have a low efficiency. However, these efficiencies may be improved if the stack gases are cooled to room temperature, by heating the feed water, for example.

5 ● What are the advantages of a sized fuel for heating furnaces?

Because a sized fuel encourages a more uniform flow of air through the bed, the burning will be more uniform, and the bed will be less liable to develop holes and will require less attention. Uniformity of fuel size is more desirable as the area of the bed becomes smaller; it is less important with fuels that cake, but with sized fuels the caking will be more uniform and the air flow through the bed will be steadier. In addition, ash and pieces of slate are less likely to be segregated and to form lumps of clinker.

6 ● Does the size of a fuel affect the quantity of air required to burn it at a given rate?

The total air required to give the same gas analysis at the stack is independent of the size of the fuel burned, but for non-caking fuels the ratio of the air passing through the fuel bed to the total air entering the burner base decreases, for the same thickness of bed, as the size of the fuel becomes smaller; this decrease is very rapid for sizes less than one inch. For coals that cake, this ratio will depend on the way the caked bed is broken up and on the size of the resulting pieces.

7 ● Is the volatile matter which is given off when coals are burned of the same nature in all coals?

No. The products given off by coals when they are heated differ materially in the ratios by weight of the gases to the oils and tars. No heavy oils or tars are given off by anthracite, and very small quantities are given off by semi-anthracite. As the volatile matter in the coal increases to as much as 40 per cent of ash-free and moisture-free coal, in-

creasing amounts of oils and tars are given up. For coals of higher volatile content, the relative quantity of oils and tars decreases, so it is low in the sub-bituminous coals and in lignite.

8 ● Is smoke a primary product in the burning of fuels?

Visible smoke may include very small particles of carbon, oil, tar, water (condensed steam), and ash. Of these, the oils, tars, and ash are mainly primary products, and the water is partly primary. The carbon, which usually comprises the greater part of the smoke, results from the breaking up by heat of oils, tars, and such gases as methane, so it may be considered a secondary product.

9 ● Is the sulphur in coals detrimental to combustion?

Not so far as is known, but its complete combustion gives only 25 per cent as much heat as is given by the same weight of carbon. Sulphur is undesirable because it causes corrosion of flues and stacks, and also because its gases pollute the atmosphere, and damage buildings and vegetation.

10 ● Can any one fuel be said to be the best fuel?

The term *best* can be applied to a fuel only after consideration of the cost factor of the fuel and the equipment necessary for its use. Gas seems to be the most convenient fuel because of its uniformity, and the ease with which it may be controlled and its burning made fully automatic.

Chapter 28

AUTOMATIC FUEL BURNING EQUIPMENT

Stokers, Residential Stokers, Apartment House Stokers, Commercial Stokers, Domestic Oil Burners, Commercial Oil Burners, Gas-Fired Appliances, Gas Boilers, Warm Air Furnaces, Space Heaters, Conversion Burners, Gas Appliances

AUTOMATIC, mechanical equipment for the efficient combustion of coal, oil, and gas is considered in this chapter.

MECHANICAL STOKERS

Assuming the same intelligence in handling the fire, coal can be burned more efficiently on a mechanical stoker than on any kind of hand-fired grate. This does not necessarily mean that a stoker installation may be more economical, because the amount of coal burned may be so small or the cost of the installation so high that the savings with stokers may not be sufficient to pay for the investment. The operation of burning coal involves uniformity in stoking, proper distribution over the fuel bed, admission of air as required to all parts of the fuel bed, and disposal of the ash. The handling of the volatile gas is largely a matter of furnace design but since this gas forms a considerable portion of the heating value of the coal, it may also be said that the proper handling of this gas is a function of firing. All mechanical stokers must provide means of taking care of these several functions in order fully to serve their purpose.

Stokers may be divided into four types according to their construction and operation, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type. They may also be classified according to their uses. The following classification has been adopted by the U. S. Department of Commerce:

Class 1. Residential (Capacity less than 100 lb coal per hour).

Class 2. Apartment houses and small commercial heating jobs (Capacity 100 to 200 lb coal per hour).

Class 3. General commerical heating and small high pressure steam plants (Capacity 200 to 300 lb coal per hour).

Class 4. Large commercial and high pressure steam plants (Capacity over 300 lb per hour).

Overfeed Flat Grate Stokers

This type is represented by the various chain grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is supplied

under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the clinker forming characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker invariably requires the use of an arch over the front of the stoker to maintain ignition of the incoming fuel and to maintain the volatile gas at a temperature suitable for combustion. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat grate stoker, but this stoker is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gas.

Underfeed Side Cleaning Stokers

In this type, the fuel is fed in at the front of the furnace to one or more retorts, is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all coking coals while in the smaller sizes it is suitable for small sizes of anthracites. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gas is released, is mixed with air, and passes through the fire where it is burned. The ash may be continuously discharged as in the small stoker or may be accumulated on a dump plate and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gas.

Underfeed Rear Cleaning Stokers

This type carries on combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed.

Class 1 Stokers, Residential

A common type of stoker in this class consists of a round retort having tuyeres at the top where all of the air for combustion is admitted. Coal is fed from a storage hopper outside of the boiler by means of a worm into the bottom of this retort and beneath the fire. The equipment includes a blower which is driven by the same motor that drives the stoker.

Some domestic stokers are provided with automatic grate shaking mechanism together with screw conveyers for removing the ash from the

ashpit and depositing it in an ash receptacle outside the boiler. Certain types can also be provided with a coal conveyer which takes coal from the storage bin and maintains a full hopper at the stoker. They may feed coal to the furnace either intermittently or with a continuous flow regulated automatically to suit conditions. Where the boiler is provided with indirect coils for heating the domestic hot water, the stoker may be so arranged that it can be used the entire year to maintain a continuous hot water supply.

Class 2 Stokers, Apartment House

This class is used extensively for heating plants in apartments and hotels, and also for small industrial plants such as laundries, bakeries, and creameries. The various stokers in this class differ materially in their design, although the majority are of the underfeed type. The principal exception is an overfeed type having step action grates in a horizontal plane and so arranged that they are alternately moving and stationary, and are designed to advance the fuel during combustion to an ash plate at the rear.

All of the stokers are provided with a coal hopper outside of the boiler. In the underfeed types, the coal feed from this hopper to the furnace may be accomplished by a continuously revolving worm or by an intermittent plunger. The drive for the coal feed may be an electric motor, or a steam or hydraulic cylinder. With an electric motor, the connection between the driver and the coal feed may be through a variable speed gear train which provides two or more speeds for the coal feed; or it may be through a simple gear train and a variable speed driver for the change in speed of the coal feed; or a simple gear train with a coal feed having an adjustment for varying the travel of the feeding device. With a steam or hydraulic cylinder, the power piston is connected directly to the coal feeding plunger.

The stokers in this class vary also in their retort design. It is customary in the worm-feed type to use a short retort in order that the unsupported length of worm within the retort may not be too weak for continuous service. In this type the retort is placed approximately in the middle of the furnace and is provided with tuyere openings at the top on all sides. In the plunger-feed type the retort extends from the inside of the front wall entirely to the rear wall or to within a short distance of the rear wall. This type of retort has tuyeres on the sides and at the rear.

This class of stokers also differs in the grate surface surrounding the retort. In many of the worm-feed stokers this grate is entirely a dead plate on which the fuel rests while combustion is completed. In the dead-plate type, all of the air for combustion is furnished by the tuyeres at the retort. Because of this, combustion is well advanced over the retort so that it may easily be completed by the air which percolates through the fuel bed. With the dead-plate type of grate the ash is removed through the fire doors and it is therefore desirable that the fuel used shall be one in which the ash is readily reduced to a clinker at the furnace temperature, in order that it may be removed with the least disturbance of the fuel bed.

In other stokers in this class, the grates outside of the retort are air-

admitting and shaking grates. These grates permit a large part of the ash to be shaken into the ash pit beneath, while the clinkers are removed through the fire doors. With this type of grate, the main air chamber extends only under the retort while the side grates receive air by natural draft from the ash pit.

In still other stokers of this class, the main air chamber extends beyond the retort and is covered with fuel-bearing, air-supplying grates. With this type of grate, the fuel is supplied with air from the main air chamber throughout combustion. Also with this type of grate, dump plates are provided beyond the grates where the ash accumulates and from which it can be dropped periodically into the ash pit beneath.

Stokers in this class are compactly built in order that they may fit into standard heating boilers and still leave room for sufficient combustion space above the grates. The height of the grate is approximately the same as that of the ordinary grates of boilers, so that it is usually possible to install such stokers with but minor changes in the existing equipment. In some districts, there are statutory regulations governing such settings.

These stokers vary in furnace dimensions from 30 in. square to approximately 66 in. square. The capacity of the stokers is measured by the amount of coal that can be burned per hour. In general, manufacturers recommend that, for continuous operation, the coal burning rate shall not exceed 25 lb of coal per square foot of grate per hour, while for short peaks this rate may be increased to 30 lb per hour. Although these stokers were designed to burn bituminous coal, they can also be used to burn the small sizes of anthracite but at a somewhat lower rate. It is often customary to have the janitor or some other attendant care for the boiler as one of his duties. Under these conditions the heating plant does not receive the same careful attention as it would if a man devoted his entire attention to the fire. With periodic hand-firing, the boiler is operated inefficiently much of the time. With a stoker, the boiler is operated at the rate that the conditions require so long as there is coal in the hopper. With hand firing, it is customary to use the more expensive sizes of fuel, while with a stoker the smaller sizes are used at a considerable saving in the cost per ton. Because the stoker responds promptly to automatic regulation, it is possible to maintain a reasonably constant standard. Also because the stoker feeds the fuel regularly and in small quantities without losses due to opening doors, it must of necessity be more efficient than hand firing. This increase in efficiency depends entirely on conditions, with a minimum of about 10 per cent and a maximum of about 25 per cent.

Class 3 Stokers, General Commercial

These stokers are suitable for the heating plants of large schools, hotels, hospitals, or other large institutions as well as industrial plants. This class is served both by overfeed stokers and by underfeed stokers. The overfeed stokers are in general of three types, (1) the chain grate, (2) the rear cleaning inclined grate, and (3) the center cleaning inclined grate or V-type.

Stokers of this type are usually operated by natural draft, although in some cases conditions permit the operation of forced draft under the

grates. With most fuels, it is not advisable to operate overfeed grates at too high a combustion rate because of the greater difficulty of cleaning and the higher maintenance, but where the fuel is free burning and has a high ash fusion temperature, the combustion rate is not so restricted. The operation of the chain grates and the rear cleaning type of inclined grates has already been described.

The V-type stoker is practically obsolete although many are still in operation. In this stoker, the grates are inclined downward from both sides of the furnace to a low point at the middle where there is either a dump plate for periodic disposal of the ash or a rotary ash grate for continuous discharge of ash. In this stoker, the fuel is fed into a hopper at the top of the grate on each side of the furnace and advanced down the grates to the center where the refuse is accumulated. This stoker is always provided with a combustion arch over the entire furnace for the purpose of assuring thorough combustion of the solid fuel and providing a furnace temperature sufficiently high to burn the volatile gases. Because of this high furnace temperature and because so little of the boiler surface is exposed to the fire to assist in carrying off the heat by radiation, this stoker is characterized by severe clinkering in the ash area. With all types of overfeed stokers, the most desirable installations are in boilers which are operated with comparatively uniform loads and moderate rates of combustion, since, even with good combustion arches, fluctuating loads or high combustion rates result in free volatile gas and this in turn means smoke.

The underfeed stokers in this class were the first of the type to be developed as at the time of their development very few large boilers were in use. The stokers are not so varied in design as those in the smaller class although in principle they are much the same. Practically all of them are of the plunger coal feed type with retorts extending the entire length of the furnace, with air supplying grates adjacent to the retorts, and with manually-operated dump plates at the sides of the furnace. The coal feeding plunger is operated by a steam or electric driver through a reduction gearing, or by a steam or hydraulic piston connected directly to the coal feeding plunger.

These stokers are heavily built and designed to operate continuously at high boiler ratings with a minimum amount of attention. Because of the fact that all volatile gas must pass through the fire before reaching the combustion chamber, these stokers will operate smokelessly under ordinary conditions. Also because of the fact that these stokers are always provided with forced draft, they are the most desirable type for fluctuating loads or high boiler ratings.

In the design of the grates for supporting the fuel between the retort and the ash plates, the stokers differ in providing for movement of the fuel during combustion. Some stokers are designed with fixed grates of sufficient angle to provide for this movement as the bed is agitated by the incoming fuel, while others have alternate moving and stationary bars in this area and provide for this movement mechanically. In either type, with proper operation, all refuse will be deposited at the dump plate. Another difference in these stokers is that some makes use a single air chamber under the whole grate area thus having the same air pressure

under the ignition area as under the rest of the grate, while others have a divided air chamber using the full air pressure under the ignition area and a reduced air pressure under the remainder of the grate. These stokers vary in size from approximately 5 ft square to a maximum of 8½ ft square.

Class 4 Stokers, Large Commercial

These stokers are usually of the underfeed type with multiple retorts and either side cleaning or rear cleaning. In the side cleaning type there may be as many as three retorts in the furnace, and the stoker functions in the same manner as has been described for the single retort. These stokers are usually limited in length to approximately 8½ ft while the width may be as great as 10½ ft. In the rear cleaning stokers the number of retorts and the dimensions of the furnace are practically unlimited.

DOMESTIC OIL BURNERS

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION
    a. *Atmospheric*—by natural chimney draft.
    b. *Mechanical*—electric-motor-driven fan or blower.
    c. *Combination of* (*a*) *and* (*b*)—primary air supply by fan or blower and secondary air supply by natural chimney draft.

2. METHOD OF OIL PREPARATION
    a. *Vaporizing*—oil distills on hot surface or in hot cracking chamber.
    b. *Atomizing*—oil broken up into minute globules.
        (1) Centrifugal—by means of rotating cup or disc.
        (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
        (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
        (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
    c. *Combination of* (*a*) *and* (*b*).

3. TYPE OF FLAME
    a. *Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
    b. *Non-luminous*—Bunsen-type flame (*i.e.*, blue flame).

4. METHODS OF IGNITION
    a. *Electric.*
        (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating or just at the beginning of operation.
        (2) Resistance—by means of hot wires or plates.
    b. *Gas.*
        (1) Continuous—pilot light of constant size.
        (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.

c. Combination—electric sparks light the gas and the gas flame ignites the oil.
d. Manual—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION
   a. On and off—burner operates only a portion of the time (intermittent).
   b. High and low—burner operates continuously but varies from a high to a low flame.
   c. Graduated—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

Air and Oil Supply

The object of regulating the air and oil supplies is to obtain a complete mixture of the proper quantities of oil and air so the fire will be clean and efficient. Proper and dependable ignition also depends upon the ability of the burner to produce consistent fuel-air mixtures. The type and shape of flame depend largely upon the methods of air and oil supply employed.

It should be pointed out that this mixture burns in a space called the furnace, which is lined with refractory bricks or other heat-resistant substances for the purpose of maintaining that space at a high temperature so that the oil and air may completely unite and burn. Excessive cooling before combustion is completed stops the combustion process and causes soot. The furnace is in some instances a valuable auxiliary in assisting in the actual mixture of oil and air and in modifying the flame shape, besides its primary function of maintaining high temperatures. The size and shape of furnace required are important, especially where the dimensions of the space into which the burner is to be placed are already fixed.

Atomization

The purpose of atomization is greatly to increase the surface area of a given quantity of oil in order to accelerate the change from the liquid state (in which oil cannot burn) to the gaseous or vaporous state, in which state it is one of the elementary fuels, gaseous hydrocarbon. This conversion is largely accomplished through the action of radiant heat energy upon the flying globules of oil, and the tremendously increased surface provided aids gasification.

Air for Combustion

Air for combustion usually is supplied by a motor-driven fan, several types being in common use. Electric motors varying from 1/20 hp to 1/3 hp are used and are started and stopped by the control mechanism. In most cases, they are direct-coupled to the fan as well as to a gear or lobe pump for drawing the oil from the storage tank, and in some cases, to a pump for forcing the oil through the nozzle.

All of the air required for combustion can be supplied by the blower, or else only the *primary* air can be supplied under pressure and provision made so that the remainder will be drawn into the combustion chamber by the natural draft developed by the chimney or by an injector-like action of the primary air. In any event there should be definite control of the quantity of air as well as of the rate of oil supply. Some method of

draft regulation is advisable in order to secure proper air regulation. It is necessary to supply more air than is actually required for complete combustion of the oil, but the amount of excess air should be reduced to the lowest workable minimum. Laboratory tests frequently show 25 per cent to 50 per cent more air than is required for combustion, yet field tests indicate that the average burner operates with from 50 per cent to 125 per cent excess air, with a corresponding reduction in the efficiency of the burner. Many domestic burners are extremes of simplicity, the only moving parts being the motor armature with a shaft and direct-connected fan and pump set.

Type of Flame, Ignition

If the vaporizing of the atomized oil and the combustion are concurrent events, a luminous flame will usually result. If vaporization and mixing are accomplished before combustion, a non-luminous flame will result. Some burners may produce either type of flame according to the adjustment made.

A limited comparison of these two types of flame shows no inherent superiority of one over the other so far as thermal efficiencies are concerned. This is definitely true when the burners are placed in boilers having ample indirect surface, and is probably true in general. The moot question of radiation has not been conclusively settled. There are indications that the radiation of luminous and non-luminous flames in boiler furnaces are practically the same. More information is needed upon this subject.

It is true, however, that a non-luminous flame may show low excess air and the presence of carbon monoxide, but no smoke. Low excess air with a luminous flame will usually show little or no carbon monoxide, but will be unmistakably smoky. Visual indications, especially with a blue flame, may therefore be quite unreliable.

When a burner is operating intermittently under the control of a thermostat, some positive form of ignition is required to function every time there is a call for heat.

The necessity for certain ignition under adverse conditions, when the line voltage is low or the oil is cold is paramount, and this phase of burner design and operation has been given the closest attention, as faulty ignition is more to be feared than improper operation once the flame is established.

The effect of the air and oil setting is important, since it may be necessary in some instances to adjust for greater excess air than is otherwise required in order to get a mixture suitable for certainty of ignition.

Recent research[1] at Yale University, conducted in coöperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*, reveals that the various methods of operation (*i.e.*, on and off, high and low, and graduated) all have potential advantages and disadvantages and that a choice in any case requires a consideration of the heat-absorbing characteristics of the boiler in which the burner is to operate. From the standpoint of efficiency of operation it seems that there is little choice if

[1]Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

at the maximum setting of all burners the boiler efficiency were at its maximum value. If, on the other hand, the boiler were operating beyond its point of maximum efficiency, then it appears that the graduated type might show better results. The following additional factors should be considered:

1. The intermittent type should be set at its point of most efficient combustion.
2. An intermittent burner should be set for a higher total heat output than either of the other two types in order to get the acceleration necessary when heat is required. This may in some cases give less economical performance due to the increased boiler load.
3. An interruption in electric current might in some instances be troublesome with continuously operating burners where manual ignition is employed.
4. The continuously operating burners must have a minimum fuel setting low enough to prevent overheating in mild weather or during the summer if the boiler is used for domestic hot water.
5. Electrical operating costs must be considered, but must be based upon known power requirements. The power requirements of some burners will be several times as high as others, so any generalization on operating costs is futile.
6. Evenness of heat supply will have some influence on uniformity of temperature.
7. Number and cost of controls which reflect in the manufacturing costs.

This entire subject, therefore, is likely to be somewhat perplexing because of the necessity of knowing, and the difficulty in determining, the efficiency characteristics of many heating boilers. Selections of oil burners on the basis of their manner of operation will probably be largely a matter of preference. The advent of special boilers for oil burning will provide the engineer with the opportunity for greater discrimination.

Temperature Control, Protective Devices

Domestic oil burners are controlled directly from the change in temperature of a designated control room (usually the living room, dining room or hall), and by temperature or pressure variations in the boiler. Oil-burner installations put in only a few years ago were simplified to the extent of having a single control element—the room thermostat—that started and stopped the burner. The modern installation provides, in addition, electrical devices inter-wired with the control system to insure against poor operation and to guard against troubles brought on by the characteristics of the heating plant.

One control system provides an instrument actuated by two temperature bulbs, one placed in the outdoor air and the other in a designated part of the heating system. The control is actuated by both bulbs and is designed to maintain the heating medium at a temperature to suit the variations in outdoor temperature, the lower the outdoor temperature the higher the temperature of the heating medium. Other devices have been developed to maintain a certain minimum temperature that will effectively prevent the downward window currents of cold air from reaching and traveling across the floor, regardless of the room thermostat.

Owing to the comparative intensity of heat production with a burner, a boiler with limited water storage above the crown sheet might pass steam to the radiator system so rapidly, at starting, that the sheet would be uncovered, with probable damage to the boiler structure. A low-water safety can be so wired into the system that the burner will be stopped before the water level is reduced to the danger point, or a boiler feed can

be installed to add water to the boiler to maintain a safe level, instead of stopping the burner. Either or both should form part of a first-class installation.

Again, with either steam or water systems, the burner control can be inter-wired with a thermostatic device having its temperature element introduced into the boiler near the top, its function being to limit the maximum temperature of water or pressure of steam so the burner will be shut off before dangerous temperatures or pressures are reached. Windows of the room in which the thermostat is located are sometimes opened to *air out* the house in the morning, and if they are not closed promptly, the burner will operate continuously and possibly develop temperature and pressure conditions that might be detrimental to the boiler. This is where the safety device can be used to offset the carelessness of the human being.

Safety controls have been developed for intermittent burners to guard against failure of ignition and in some instances against momentary flame failures. In general, regulatory devices are well developed and dependable. Otherwise the domestic oil burner probably would not have been possible.

For further information on temperature control with oil burners, see Chapter 14.

Boilers for Domestic Oil Burners[2]

Boilers used with domestic installations may be those designed for solid fuel or those designed for liquid fuel. The latter are coming to the fore with great rapidity as they usually have greatly increased secondary surface. Many are of copper or steel tube design. Increased efficiencies of 5 to 15 per cent are often obtainable with boilers designed especially for liquid fuel.

It is possible to go to extremes in providing secondary surfaces sufficient to reduce flue temperatures to the order of 250 F to 300 F, with the result that the added resistance through the flues may necessitate the use of a booster fan to insure sufficient draft. It is difficult to obtain satisfactory efficiencies with boilers having little or no secondary surfaces, where the hot products of combustion pass almost immediately from the combustion chamber to the flue; in fact a high efficiency is unlikely with any fuel under such conditions, and the intermittent burner is especially at a disadvantage because of its characteristic development of heat at a high rate while it is operating.

It is essential that the flame produced by an oil burner, especially where it is strongly luminous, be kept from contact with the water-backed surfaces of the combustion chamber, and to this end bricking or its equivalent must be provided in most cases. Where a burner fires through the ash pit, doorframe bricking must protect the unbacked surfaces of the ash pit. The same fire bricking constitutes the actual combustion chamber for the burner flame, and materially increases the combustion volume for a given boiler.

[2]For additional information on this subject, refer to Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

Installation

The intelligence and care with which a burner is installed largely determine the satisfaction that will result from its operation. Two plans for the installation of burners are in general use. In the first, the dealer makes all installations. In the other, sales agencies function only to make sales, and the installation for as many as twenty such sales offices is done by a centrally located installation force, usually factory controlled.

Some burners are adjusted for oil rate by means of a blind needle valve that can be operated only with a special wrench; others, by changing the size of the orifice; others, by a combination of orifice size and pressure. In any event, changes in the firing rate, involving careful air and draft adjustment to match the oil rate, should be made by only a trained man,

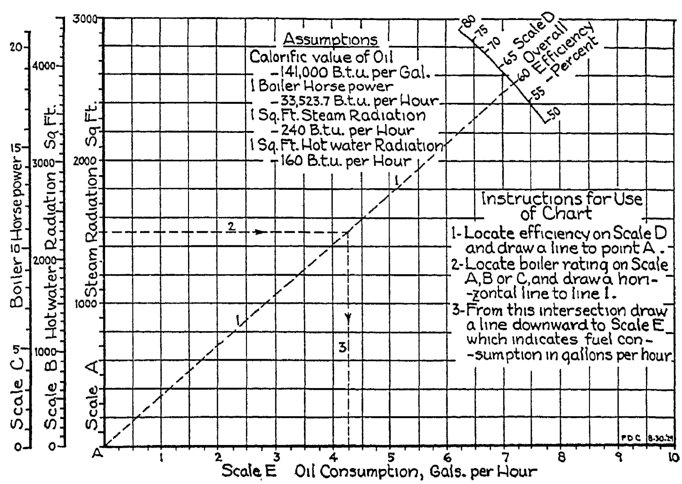


FIG. 1. FULL LOAD RATE OF OIL CONSUMPTION FOR HEATING BOILERS

preferably with the aid of an Orsat test set so that the degree of combustion efficiency can be determined. It is practically impossible to set a burner flame by eye, although that has been general practice in the past. The industry is turning to the Orsat and, as a result, more domestic burners are operating at from 9 to 12 per cent CO_2, representing a higher efficiency combustion than at 5 to 8 per cent, as frequently is the case where the burner is adjusted by eye.

Air for Combustion

It is essential that the basement, or at least that portion used as a boiler room, be open to the outside air, in order that sufficient air be available for combustion. Frequently a case of poor operation will be found where a test with a draft gage made by inserting the tube through the keyhole of the outer door will show that there is a partial vacuum in the basement when the burner is running, all of the combustion air coming

through the keyhole and minute cracks. A simple remedy is to cut an inch from the bottom of the outer door.

In order to achieve satisfactory heating at the lowest cost, careful consideration should be given to oil, air and draft adjustment. The oil adjustment should be determined from the total heat requirements to be met. The heat loss of the building plus an allowance for piping plus 20 to 25 per cent for pick-up establishes the maximum output required from the boiler. Fig. 1 indicates the oil required in gallons. Piping allowances will usually vary between 25 and 10 per cent, decreasing with an increase in the size of the building.

With the oil rate thus fixed, the air and draft should be set to give efficient combustion (that is, 10 to 12 per cent CO_2). The furnace draft should be set reasonably low and should be maintained constant by means of an automatic draft regulator. Without this the air supply will fluctuate, causing uneven performance. A check should be made to insure that ignition will be satisfactory under all conditions. An oil burner of the continuous type might dispense with all or part of the pick-up allowance due to the nature of its operation. Careful adjustment will provide ample heat output under all conditions, will minimize the load on the boiler, and will establish the most favorable conditions for intermittent operation.

An essential element in the satisfactory operation of domestic oil burners is the provision for maintenance and service for the burners. What might be called *emergency service* for mechanical or electrical failure of the burner has rapidly diminished during the last few years until a level has been reached where groups of 100 to 1000 burners in a community consistently will require an average of not more than one call per burner per heating season. Maintenance service is coming into general practice where, for a fixed annual payment, regular inspection is made of the burner, and faulty operation corrected before the burner becomes inoperative. This service may contemplate entire overhauling of the burner during the summer, and may include annual cleaning of the boiler flues with a specially designed vacuum cleaner.

Domestic Hot Water Supply

Provision may be made for heating domestic water through exchange heaters attached to the boiler, in which water is maintained at a fixed temperature or steam at a set pressure during the entire year. The flow of water or steam to the radiators is controlled by electrically-operated valves, which remain closed during warm weather and open (through the functioning of the room thermostat) when heat is required in the house. The room thermostat either causes heat to be produced by starting the burner when the room temperature drops to a predetermined point, or closes the circuit of the motor by operating a valve in the flow line of the heating system, the motor opening the water or steam valve and permitting water or steam immediately to flow to the radiators. When the flow in a water heating system is sluggish, the room thermostat also can start the motor of a circulation pump, thereby decreasing the time required to bring the room temperature up to the desired point.

It is usual in small steam heating systems to dispense with the motor-

operated valve and by means of an aquastat maintain the boiler water at a constant temperature but well below the steaming temperature (*i.e.*, 140 to 180 F). The lowest temperature setting that will produce sufficiently hot water will be the most economical. The aquastat will always function in such a way as to maintain this temperature except when the room thermostat calls for heat, which means that a call for steam can be more quickly obtained.

Another type of control valve available for hot water systems is thermostatically operated so as to prevent a flow of water to the heating system until the call of the room thermostat for heat raises the water temperature above that normally required for domestic hot water. It should be noted that, except in the case of the graduated burner, the water temperature in the heating system will nearly always reach its maximum, thereby depriving this system to some degree of its natural advantage of modulation.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups producing a horizontal torch-like flame. As much as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler, and heating the oil to within 30 deg of its flash point.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ash-pit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2 lb of oil can properly be burned. This corresponds to a maximum liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. This in turn suggests that the value of 38,000 Btu per cubic foot per hour might be adjusted according to good engineering judgment. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire-brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas-heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

I. Gas-Designed Heating Systems.
   A. Central Heating Plants.
      1. Steam, hot water, and vapor boilers.
      2. Warm air furnaces.
   B. Unit Heating Systems.
      1. Warm air floor furnaces.
      2. Industrial unit heaters.
      3. Space heaters.
      4. Garage heaters.

II. Conversion Heating Systems.
   A. Central Heating Plants.
      1. Steam, hot water and vapor boilers.
      2. Warm air basement furnaces.

The majority of these systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push button control, or manual control.

Although no exact rules can be prescribed as to the field best covered by each of the foregoing systems, each installation will have problems pointing more or less directly to some particular type of heating equipment.

Gas-Fired Boilers

Information on gas-fired boilers will be found in Chapter 25.

Either snap action or throttling control is available for gas boiler operation. This is especially advantageous in straight steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Warm Air Furnaces

There are two general classes of gas-fired warm air furnaces, the gravity furnace which depends upon the natural tendency of heated air to rise, providing the proper circulation of heated air into the room, and the mechanical circulation furnace by which the air to be heated is forced through or drawn through the furnace by means of a fan.

Warm air furnaces are variously constructed of cast iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates, such as that of southern California. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar as far as the period of heating and temperature to be maintained are concerned. Bedrooms, living rooms, and dining rooms often present excellent possibilities for this type of furnace.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas up into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces. The desirability of using an appliance designed for gas, when gas is to be the fuel, applies even more strongly to furnaces than to boilers.

Codes for proportioning warm air heating plants, such as that formu-

lated by the *National Warm Air Heating Association* (see note p. 401), are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coördinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Floor Furnaces

Warm air floor furnaces are well adapted for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below. Air is taken downward between the two sheets of the double casing and discharged upward over the heating surfaces and into the room. The appliance is controlled from the room to be heated by means of a control lever located near the edge of the register. The handle of the control is removable as a precaution against accidental turning on or off of the gas to the furnace.

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. Space heaters are quite extensively used for house heating in milder climates such as exist in the South and Southwest. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although they should be connected with solid piping it is sometimes desirable to connect them with flexible gas tubing in which case a gas shut-off on the heater is not permitted, and only *A.G.A.* approved tubing should be used.

Space Heaters

Parlor furnaces or *circulators* are usually constructed to resemble a cabinet radio. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass up around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters make admirable auxiliary heating appliances to be used during the occasional cool days at the beginning and end of the heating season when heat is desired in some particular room for an hour or two. The radiant heater gives off a considerable portion of its heat in the form of radiant energy emitted by an incandescent refractory that is heated by

a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to the ordinary wall register. Others are encased in frames which fit into fireplaces.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator connected to a basement boiler. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to the steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged out a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system, especially in natural gas territories. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type with luminous flames, operating without refractories. In each case an attempt is made to transfer the majority of the heat from the gas to the medium to be heated within the fire pot itself because of the low heat transfer that takes place in the flue passages.

Many conversion units are equipped with sheet metal secondary air ducts which are inserted through the ash-pit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. By means of this duct the air necessary for proper combustion is supplied directly to the burner, thereby making

it possible to reduce the amount of excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push button, or room temperature control.

Sizing Gas-Fired Heating Plants

While gas-burning equipment can be and usually is so installed as to be completely automatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a

TABLE 1. SELECTION FACTORS FOR GAS BOILERS

CAST-IRON STEAM RADIATION (EQUIVALENT SQUARE FEET)	SELECTION FACTOR (PER CENT)
500	56.0
800	54.0
1,200	51.0
1,600	48.0
2,000	45.0
3,000	42.5
4,000 and over	40.0

much lower selection, or safety, factor. A gas-fired boiler under thermostatic control is so sensitive to variations in room temperatures that in most cases a factor of 25 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Consequently, a selection factor for thermostatically controlled boilers must be variable. Table 1 gives selection factors to be added to the installed steam radiation under thermostatic control. They have been established by experience and are recommended by the *American Gas Association.*

The same factors may be used in determining the gas demand for which conversion burners installed in steam or hot water boilers should be set. Multiplying the equivalent direct heating surface (radiation) by 240 and adding the appropriate percentage from Table 1, and then dividing by the heat value of the gas and by the heating efficiency (see discussion of heating efficiencies in Chapter 29), gives the proper hourly rate of gas consumption. However, inadequate boiler heating surface for gas burning, often encountered in coal-designed boilers converted to gas, may necessitate operation at a lesser demand, resulting in much slower pick-up and less margin of safety for piping loss.

Appliances used for heating with gas should bear the approval seal

of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150[3] for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. This same limitation applies to all classes of gas-consuming heating appliances that are tested and approved by the Laboratory. Gas boilers are available with ratings up to 14,000 sq ft of steam, while furnaces with ratings up to about 500,000 Btu per hour are available. (See Chapter 23.)

Installation Features

One feature of the piping installation that adds to the satisfactory service rendered by gas boilers is provision for adequate and rapid venting of the air from steam heating systems. If air leaks into the steam distribution system during the period that the gas is turned off, and then vents out slowly when the thermostat calls for heat, the result will be a further cooling of the premises between the time that the thermostat calls for heat and the time that steam reaches the radiators. A freely venting steam or vapor system gives maximum economy and minimum temperature variation. When gas boilers are attached to existing heating plants, it is good practice to check the effectiveness of the venting devices

[3]A value of 160 for the heat emission of hot water radiators is used by many engineers. The actual heat emission, however, depends on the temperature of the water and of the surrounding air. See Chapters 30 and 33.

and if necessary to replace them with more effective ones that will prevent the return of air into the heating system, and also to check the tightness of the piping.

Frequently when a coal boiler is already installed in a home, it is expedient to leave the coal boiler in place, and to cross-connect the gas boiler with it. Where gas heating is new to the community, it produces a more secure feeling in the customer's mind when putting in gas-fired house-heating equipment, if he knows that he can burn coal at any time he may desire. For steam or vapor installations, it is desirable to have the water line in both boilers at the same level.

PROBLEMS IN PRACTICE

1 ● What functions must an automatic stoker perform in burning coal?

An automatic stoker must distribute the coal evenly over the fuel bed, and fire it uniformly. It must introduce air in proper quantities to all parts of the fuel bed, and dispose of the ash without interfering with the combustion process. Indirectly, the stoker is responsible for the proper burning of the volatile fuel gases in the combustion space above the fuel bed.

2 ● Classify stokers as to construction and operation.

a. Overfeed flat grate.

b. Overfeed inclined grate.

c. Underfeed side cleaning type.

d. Underfeed rear cleaning type.

3 ● What classification may be made of stokers as to their use?

Class 1. For residences (Capacity less than 100 lb of coal per hour).

Class 2. For apartment houses and small commercial heating jobs (Capacity 100 to 200 lb of coal per hour).

Class 3. For general commercial heating and small high pressure steam plants (Capacity 200 to 300 lb of coal per hour).

Class 4. For large commercial and high pressure steam plants (Capacity over 300 lb of coal per hour).

4 ● What main parts are found in an underfeed residential stoker?

A *hopper* is supplied to hold coal which is fed by a *screw* or *plunger* into a *retort* provided with air openings called *tuyeres*. A *blower* supplies air under pressure for combustion, and a *gear case* provides for changes in coal feeding rates.

5 ● What is a dead-plate?

A dead-plate is a flat surface without air supply openings upon which the fuel rests while combustion of the fixed carbon is completed. Generally the ash is removed from the dead-plate.

6 ● What rate of coal burning is usually recommended for small underfeed stokers?

For continuous operation, 25 lb per square foot of grate surface is recommended; for short duration peaks, 30 lb.

7 ● What methods of oil atomization are used?

1. Throwing the oil from a rotating cup or disc.
2. Forcing the oil under high pressure through a whirl chamber in a nozzle.
3. Propelling the oil with a high velocity jet of air or steam.
4. Forcing an oil and air mixture through a nozzle.

8 ● What is the purpose of atomization?

Atomization is used to increase the surface area of the oil in order to facilitate putting it into a vaporous state so it may burn.

9 ● Is the furnace of much importance in oil burning?

In most cases it is very important. It is the function of the oil burner to supply the air and fuel in correct proportions; the furnace must provide heated space for proper mixing and combustion.

10 ● Which flame is considered better, the luminous or the non-luminous?

Laboratory tests show that they are equally efficient in the usual installation.

11 ● What main precaution is necessary in choosing a boiler for an oil burner?

Since the burner output is usually varied through a wide range under control of the thermostat, a boiler should be provided with enough indirect heating surface to absorb the heat as it is released. The combustion space must be large enough, and have correct proportions for mixing fuel and air at high temperatures. If oil is used inefficiently high heating costs will result.

12 ● How should oil burner adjustments be made?

Adjustments should be made by an experienced man who uses a gas analysis apparatus to determine the CO_2 content.

13 ● What CO_2 content should be attained in oil burning?

Ten per cent CO_2 is considered good practice, for it indicates the supplying of 50 per cent excess air.

14 ● What maximum heat release is considered good practice in oil burning?

A heat release of 38,000 Btu per cubic foot per hour is considered to be the maximum for average large installations. This figure has been greatly exceeded in some cases. The design of the combustion chamber, as to impingement of flame and as to proper mixing at high temperatures, has much to do with the attainable heat release.

15 ● Name five types of gas-fired space heaters.

a. Parlor furnaces or circulators.
b. Radiant heaters.
c. Gas-fired steam or hot water radiators.
d. Warm air radiators.
e. Garage heaters.

16 ● How are gas heating units rated?

Gas-fired units are rated on the basis of output in Btu per hour.

17 ● What safety consideration is noted in establishing the ratings of gas-fired units?

The rating is limited by the amount of gas that can be burned without the liberation of harmful amounts of carbon monoxide.

18 ● What control equipment is essential in the usual oil burner installation?

See Chapter 14. Usually a room thermostat, a limiting device to prevent the pressure or temperature from exceeding a desirable limit, a shut-off device to guard against failure of flame ignition, and in steam or hot water boilers a low water protective device.

19 ● In the construction of gas-fired garage space heaters, what special precaution must be taken?

A safety screen must be placed over each opening into the combustion chamber to prevent explosions of any possible gasoline vapors.

20 ● List some factors which might account for possible economies of stoker firing over hand firing.

a. The regular feed of coal instead of the intermittent feed.

b. The use of cheaper sizes and grades of fuel.

c. The absence of door openings for firing purposes, which avoids the admission of cold excess air.

d. The avoidance of overheating because the stoker responds quickly to automatic equipment controlled by the heat demand.

Chapter 29

FUEL UTILIZATION

Heat Loss, Calorific Values, Heating Efficiencies, Non-Heating Periods, Heat Capacity of Buildings, Miscellaneous Factors, Degree-Day Method, Rough Approximations, Relative Heating Costs

TO predict the amount of fuel likely to be consumed in heating a building during a normal heating season, it is necessary to know the total heat requirements of the building and the utilization factor of the fuel. The accuracy of the estimate will depend on the ability to select these values and on the care taken in making allowances for other variable factors.

Fuel requirements[1] are given by the following general equation:

$$F = \frac{H \times (t - t_a) \times N}{(t - t_o) \times C \times E} \tag{1}$$

where

F = quantity of fuel required for a heating season.
N = number of hours of heating season.
t = inside temperature, degrees Fahrenheit.
t_a = average outside temperature, degrees Fahrenheit.
t_o = outside design temperature, degrees Fahrenheit.
H = calculated heat loss of building based on outside temperature of t_o, Btu per hour.
C = calorific value of one unit of fuel, the unit being the same as that on which F is based.
E = efficiency of utilization of fuel, per cent.

HEAT LOSS

The hourly heat loss (H) is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i) of the rooms or spaces to be heated, and the total equivalent heating surface required is equal to $\frac{H}{240}$ sq ft. In estimating the fuel consumption of a building of more than one room divided by walls or partitions, it is not correct to use the calculated heat loss of the building without making the proper allowance for the fact that the heating load at any time does not involve the sum of the infiltration losses of all of the heated spaces of the building but only part of the infiltration losses. This is explained in Chapter 6.

It is sufficiently accurate in most cases to consider only half of the total infiltration losses of a building having interior walls and partitions, and the value of H in Equation 1 would, under these conditions, be equal to

[1]For further information on this subject see Estimating Fuel Consumption, by Paul D. Close, (*Heating Piping and Air Conditioning*, May, 1931).

$H_t + \frac{H_i}{2}$. If a building has no interior walls or partitions, whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side, and only half of the total crack should be used in computing the infiltration for each side and end of the building. Under these conditions it is sufficiently accurate to use the total calculated heat loss (H) for the building. If the average wind velocity during the heating season differs from that upon which H_i was derived, the value of H should be corrected accordingly.

Of course, where the required heating surface is estimated by empirical or rule-of-thumb methods, refinements in approximating fuel consumption are not warranted, but rule-of-thumb methods often lead to unsatisfactory results and should be avoided in heating work where more accurate methods are available. It should be emphasized that the value of H in Equation 1 is the total heat loss of the building after making the proper allowance for infiltration.

CALORIFIC VALUES AND HEATING EFFICIENCIES

The calorific values of fuel oils and gas can be ascertained with reasonable accuracy. The values for various grades of oil are given in Table 3, Chapter 27. The calorific value of gas can always be obtained from the local utility company. Values for natural gas are given in Table 4, Chapter 27; manufactured gas usually has a calorific value of about 535. Coals have a larger range and may vary for the same type of coal, depending on its ash content. For general purposes where specific data are lacking, values can be taken from Table 1, Chapter 27.

To decide on the correct efficiency to use is a more difficult matter, particularly if the estimate is being made without a full knowledge of the equipment for burning the fuel and the care the furnace will receive. Efficiencies usually are given in the catalogs of manufacturers of furnaces and boilers, but these values are obtained under test conditions and do not allow for poor attendance, defects in installation, or poor draft. On the other hand, such efficiencies assume that all the heat radiated from the outside of the heaters or casings as sensible heat of the flue gases is lost, whereas, if the heater is installed in the building being heated, a considerable portion of these losses may help to heat the building[2]; how much of this it is legitimate to use in increasing the value of E will depend on whether H included the heat losses in the cellar, and on the construction of the chimney. Except for an interior chimney, the heat transferred through the chimney wall to the building will be very small. Chimney allowances should be greater for lower test efficiencies. Thus an insulated furnace will give a high efficiency on test but will not heat the cellar. A modern gas furnace will have a high efficiency with a correspondingly low flue gas temperature and hence there will be very little heat from the flue pipe.

For great exactitude the value for E should take care of inefficiency in the heat distribution in the building because of such losses as excessive heating of the walls behind the radiators and excessive stratification. It

[2]Analysis of the Over-All Efficiency of a Residence Heated by Warm Air, by A. P. Kratz and J. F. Quereau (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

is preferable, however, to include these losses in the value of H, and to limit E to the fuel burning equipment.

Automatic fuel burning equipment, whether for coal, oil or gas, will tend to save fuel and will therefore produce a higher efficiency if thermostatically controlled, but on the other hand automatic equipment tends to make the householder prolong his heating season and maintain a higher temperature in the house in the early fall and late spring.

NON-HEATING PERIODS

Obviously, the theoretical fuel consumption will be reduced considerably by not operating the heating plant at night. Allowance for this may be made in either of two ways: (1) by estimating the average inside temperature t, or (2) by arbitrarily assuming a certain reduction in the fuel consumption.

The first procedure is, of course, the more accurate. If, for example, the daytime temperature is to be 70 F, and the temperature from 12 midnight to 6 a.m. is to be maintained by thermostatic control at 50 F, then the average daily inside temperature t will be $\frac{18 \times 70 + 6 \times 50}{24}$ or 65 F. Strictly speaking, this average inside temperature would apply only when the outside night temperature averages below 50 F, but this fact usually is not of sufficient importance to warrant consideration. If the average outside temperature during the heating season is 30 F, the fuel saving would be approximately $100 \times \frac{70 - 65}{70 - 30}$ or 12.5 per cent. In this case, the additional saving in fuel due to the cooling of the air and structural materials to 50 F would be offset by the heating-up load in the morning.

As to the second procedure, it may be arbitrarily assumed that a saving in the fuel consumption of from 10 to 30 per cent, depending on conditions, will result if the heat is shut off after working hours, and the building is heated to the required temperature during the period of occupancy each day. This, of course, is a general statement and wherever possible the average temperature should be estimated from the proportionate lengths of the occupancy and non-occupancy periods and the corresponding temperatures for these periods. Any deviation from the assumed inside temperature will result in a variation in the estimated fuel consumption.

HEAT CAPACITY OF BUILDINGS

The heat required to warm the cold building and contents is a factor to be considered. Under certain conditions, the cooling of the structure and contents will, to some extent, compensate for the heat required to rewarm the building. For example, if the building is under thermostatic control and the day and night temperatures are say 70 F and 50 F, respectively, there will be a period during which no heat will be called for while the building is cooling to 50 F, and the saving resulting therefrom will correspond to the additional heat required to bring the building and contents back to the daytime temperature. If in estimating the fuel consumption the average daily inside temperature is based on the proper day and night temperatures and periods, the heat required to warm the structure may be neglected.

Where irregular conditions are involved it may be desirable to actually calculate the fuel required to warm the building structure and contents for the number of times during the heating season the heating plant would not be in operation and to add this quantity to the fuel required for the number of hours during which the building is heated. The greater the heat capacity of the structure the greater will be the relative importance of this item. For structures of low heat capacity, such as frame buildings, this factor usually may be neglected.

Example 1. A small factory building located in Philadelphia is to be heated to 60 F between the hours of 7 a.m. and 7 p.m., and to 50 F during the remaining hours. The calculated hourly heat loss, based on a design temperature of − 6 F, is 500,000 Btu. If coal having a calorific value of 12,500 Btu per pound is fired, and the over-all heating efficiency is assumed to be 60 per cent, how many tons of coal will be required for a normal heating season, neglecting other heat sources and any loss of heat through open windows?

Solution. Since there are no partitions in this building, the entire heat loss is considered. The average outside temperature during the heating season t_a is 41.9 F (see Table 2, Chapter 7); $t = 60$ F; $N = 5040$; $H = 500{,}000$; $(t - t_o) = 66$ F; $C = 12{,}500$; $E = 0.60$. Substituting these values in Equation 1 and dividing by 2000 to change to tons:

$$F = \frac{500{,}000 \times 18.1 \times 5040}{66 \times 12{,}500 \times 0.60 \times 2000} = 46 \text{ tons of coal}$$

Inasmuch as the building will be heated to 50 F at night, the average inside temperature at the breathing line will be 55 F, and the percentage saving will be $\frac{60 - 55}{60 - 41.9}$ = 0.276 or 27.6 per cent. The net fuel consumption will therefore be 46 − 0.276 × 46 or 33.3 tons.

MISCELLANEOUS FACTORS

There are many factors which would be likely to affect the theoretical fuel requirements of a building, such as the opening of windows, abnormal inside temperatures, other heat sources, sun effect, wind, and rain. In many cases it is difficult to evaluate these factors accurately, particularly in the case of open windows, and the results are correspondingly less accurate. The degree of refinement of the calculations should, of course, be consistent with the conditions involved. If the heat loss from the boiler and piping does not warm the building or is not included in H, the proper allowance should be made. In selecting a boiler, this allowance is frequently assumed to be 25 per cent of the total heat loss of the building, but in estimating fuel requirements, the more accurate procedure of computing the pipe and boiler losses should be used, unless this item is likely to be outweighed by other less tangible factors.

Where temperature control is installed the fuel consumption can obviously be predetermined with greater accuracy than where no such control has been provided. In fact the calculated requirements agree to a remarkable extent in many cases with the actual fuel consumption. This has been particularly true of gas-fired installations, with which effective temperature regulation usually is possible.

OTHER HEAT SOURCES

Where other heat sources are available it is quite often possible to make accurate allowance for the reduction in the fuel consumption resulting

therefrom. These sources include the heat supplied by persons, lights, motors and machinery, and should also be ascertained in the case of theaters, assembly halls and industrial plants. (See Chapter 7.) In many cases these heat sources should not be allowed to affect the size of the installation of heating equipment, although they may have a marked effect upon the fuel consumption. In residences this factor usually may be neglected.

DEGREE-DAY METHOD

A very useful unit for estimating fuel consumption, particularly for residences, is the degree-day. (See definition in Chapter 41.) Degree-days for various cities in the United States and Canada are given in Table 1. The term degree-day originated in the gas industry and was later standardized by the *American Gas Association*[3].

The base of 65 F is used for an inside temperature of 70 F. This base was chosen because it was demonstrated, by means of data collected from numerous installations, that heat is seldom supplied to a residence when the outdoor temperature is greater than 65 F. It was also found that the fuel consumed varied almost directly with the difference between 65 F and the outside temperature.

If the inside temperature were maintained at 70 F throughout the 24 hours of the day, then the base of 65 F would probably be in error. It must be borne in mind, however, that although the temperature head is the difference between the inside temperature of say 70 F, and the outside temperature, a lower temperature than 70 F will usually be maintained at night and the base of 65 F will therefore allow for this condition. As already indicated, a temperature of 50 F from midnight to 6 a.m. will reduce the 24-hour average from 70 to 65 F. It is important to note that the degree-day applies specifically to an inside temperature of 70 F, which is the usual temperature for residences, and it should also be noted that allowance is automatically made for the lower nighttime temperature, although this allowance is constant for any given locality.

In Equation 1, the quantity $(t - t_a) \times N$ is equivalent to the number of degree-days D in a heating season multiplied by 24, when the average daily value of t is 65 F. Therefore,

$$(t - t_a) \times N = 24\,D \tag{2}$$

Substituting the value of $(t - t_a) \times N$ from Equation 2 in Equation 1, the following general formula for an average daily inside temperature of 65 F, which is approximately equivalent to an inside daytime temperature of 70 F for residences, is obtained:

$$F_d = \frac{24\,HD}{(t - t_o) \times C \times E} \tag{3}$$

Example 2. The calculated hourly heat loss of a residence located in Chicago is 127,000 Btu, which includes 28,000 Btu for infiltration. The design temperatures are − 8 F and 70 F. The normal heating season is assumed to be 210 days (5,040 hours) and the average temperature during this period is 36.4 F (see Table 2, Chapter 7). The

[3]See *Industrial Gas Series, House Heating (third edition)* published by the *American Gas Association*.

TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

Col. A State	Col. B City	Col C Degree-Days	Col. A State	Col B City	Col. C Degree-Days
Ala.	Birmingham	2,408	Nev.	Reno	5,891
	Mobile	1,471	N. H.	Concord	6,852
Ariz.	Flagstaff	7,145	N. J.	Atlantic City	5,175
	Tucson	1,845		Trenton	4,934
Ark.	Hot Springs	2,665	N. M.	Santa Fe	6,063
	Little Rock	2,811	N. Y.	Albany	6,889
Calif.	Los Angeles	1,504		Buffalo	6,821
	San Francisco	3,264		New York	5,348
Colo.	Colorado Springs	6,553	N. C.	Raleigh	3,234
	Denver	5,873		Wilmington	2,302
Conn.	New Haven	5,895	N. Dak.	Bismarck	8,498
D. C.	Washington	4,626	Ohio	Cincinnati	4,702
Fla.	Jacksonville	890		Cleveland	6,154
Ga.	Atlanta	2,891		Columbus	5,323
	Savannah	1,490	Okla.	Oklahoma City	3,613
Idaho	Boise	4,558	Ore.	Portland	4,468
	Lewiston	4,924		Salem	4,629
Ill.	Chicago	6,315	Pa.	Philadelphia	4,855
	Springfield	5,370		Pittsburgh	5,235
Ind.	Evansville	4,164	R. I.	Providence	6,014
	Indianapolis	5,297	S. C.	Charleston	1,769
Iowa	Des Moines	6,373		Spartanburg	3,257
	Sioux City	7,023	S. Dak.	Sioux Falls	7,683
Kans.	Dodge City	5,034	Tenn.	Memphis	2,950
	Topeka	5,301		Nashville	3,578
Ky.	Lexington	4,616	Texas	Austin	1,578
	Louisville	4,180		Dallas	2,455
La.	New Orleans	1,023		Houston	1,157
Me.	Eastport	8,531		San Antonio	1,202
	Portland	7,012	Utah	Logan	6,735
Md.	Baltimore	4,333		Salt Lake City	5,553
Mass.	Springfield	6,464	Vt.	Burlington	7,620
	Boston	6,145	Va.	Fredericksburg	4,243
Mich.	Detroit	6,494		Norfolk	3,349
	Marquette	8,692		Richmond	3,725
Minn.	Duluth	9,480	Wash.	Seattle	4,868
	Minneapolis	7,851		Spokane	6,353
Miss.	Vicksburg	1,822	W. Va.	Morgantown	5,016
Mo.	Kansas City	5,202		Parkersburg	4,884
	St. Louis	4,585	Wis.	Fond du Lac	7,612
Mont.	Billings	7,115		Green Bay	7,823
	Havre	8,699		La Crosse	6,690
Nebr.	Lincoln	6,231		Milwaukee	7,372
	Omaha	6,128	Wyo.	Cheyenne	7,462

Province	City	Degree-Days	Province	City	Degree-Days
B. C.	Victoria	5,777	Ont.	Toronto	7,732
	Vancouver	5,976	Que.	Montreal	8,705
	Kamloops	6,724		Quebec	8,628
Alb.	Medicine Hat	8,152	N. B.	Fredericton	9,099
Sask.	Qu'Appelle	11,261	N. S.	Yarmouth	7,694
Man.	Winnipeg	11,166	P. E. I.	Charlottetown	8,485
Ont.	Port Arthur	10,803			

[a]From *Industrial Gas Series, House Heating (third edition)* published by the *American Gas Association.* These degree-days are based on daily mean temperatures. Base, 65 F.

building is to be heated with oil fuel having a calorific value of 141,000 Btu per gallon. The heating efficiency is assumed to be 70 per cent. Thermostatic control is to be used and a temperature of 55 F is to be maintained from 11 p.m. to 7 a.m. How many gallons of oil will be required during a normal heating season if the loss of heat through open windows is neglected?

Solution. The maximum hourly heat loss will be $127{,}000 - \frac{28{,}000}{2} = 113{,}000$ Btu $= H$. Substituting the proper values in Equation 1:

$$F = \frac{113{,}000 \times (70 - 36.4) \times 5040}{141{,}000 \times 0.70 \times [70 - (-8)]} = 2486 \text{ gal of oil.}$$

The average inside temperature will be $\frac{70 \times 16 + 55 \times 8}{24} = 65$ F

and the fuel saving due to this fact will be $\frac{70 - 65}{70 - 36.4} = 0.149$ or 14.9 per cent.

Hence, the net fuel consumption will be $2486 - 0.149 \times 2486 = 2116$ gal.

The normal number of degree-days for Chicago is 6315. Substituting in Equation 3 and solving by the degree-day method:

$$F = \frac{113{,}000 \times 6315 \times 24}{78 \times 141{,}000 \times 0.70} = 2225 \text{ gal of oil}$$

No allowance need be made for the average temperature of 65 F since this is taken care of by the selection of a base of 65 F for the degree-day, as already explained. It will be noted that the two methods check within 5 per cent in this case. If the average daily inside temperature in the first solution had been 66.4 F instead of 65 F, the two methods would have checked exactly.

INDUSTRIAL DEGREE-DAY

Since the standard degree-day is intended for an inside temperature of 70 F, it is particularly convenient for solving residence problems. Where the design temperature differs greatly from 70 F, the standard degree-day cannot be accurately applied. Consequently, the industrial degree-day[4] has been developed and values have been derived for two bases, namely 55 F and 45 F, intended for inside temperatures of 60 F and 50 F, respectively.

There is a considerable spread, however, among these three bases, and consequently there would be an appreciable error if the actual basis to be used in a certain case would be approximately midway between any two of the three bases for which degree-day values are at present available. Since the correction cannot be made on a proportionate basis, it would be more accurate in the majority of cases involving inside temperatures other than 70 F, 60 F, or 50 F to apply Equation 1.

APPROXIMATING FUEL REQUIREMENTS

It is sometimes desirable to obtain a rough approximation of the annual fuel consumption. Such approximations may be obtained by using unit factors based on the fuel requirements per square foot (or per 100 sq ft) of *radiation* or per 1000 cu ft of space.

Fig. 1 may be used for rough approximations of coal and oil require-

[4]See *Heating and Ventilating* Degree-Day Handbook.

ments. It should be noted that this figure is given in terms of the fuel consumption per 1000 degree-days per 100 sq ft of equivalent heating surface (steam) based on an emission of 240 Btu per square foot. Unless the *radiation* is calculated with reasonable accuracy, unit factors will be of little value even for rough approximations, since it is obvious that such *radiation* requirements must bear some relationship to the actual heating requirements of the building.

Example 3. Estimate the approximate coal consumption for a building located in New York City in which the calculated heating surface requirements (steam) are 1000 sq ft based on design temperatures of zero and 70 F.

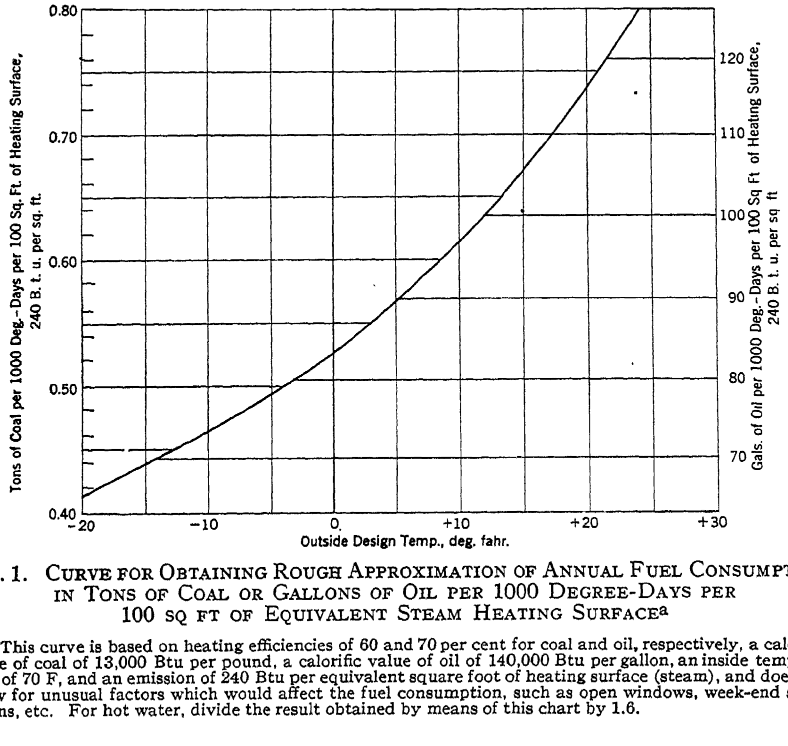


FIG. 1. CURVE FOR OBTAINING ROUGH APPROXIMATION OF ANNUAL FUEL CONSUMPTION IN TONS OF COAL OR GALLONS OF OIL PER 1000 DEGREE-DAYS PER 100 SQ FT OF EQUIVALENT STEAM HEATING SURFACE[a]

[a]This curve is based on heating efficiencies of 60 and 70 per cent for coal and oil, respectively, a calorific value of coal of 13,000 Btu per pound, a calorific value of oil of 140,000 Btu per gallon, an inside temperature of 70 F, and an emission of 240 Btu per equivalent square foot of heating surface (steam), and does not allow for unusual factors which would affect the fuel consumption, such as open windows, week-end shutdowns, etc. For hot water, divide the result obtained by means of this chart by 1.6.

Solution. From Fig. 1, the fuel consumption for a design temperature of zero is 0.53 tons per 1000 degree-days per 100 sq ft of heating surface. Since there are 5348 degree-days in New York City in a normal heating season, the fuel consumption will be approximately $0.53 \times 5.348 \times 10 = 28.34$ tons.

Fig. 2 is taken from the 3rd edition of Industrial Gas Series on House Heating, published by the *American Gas Association*, and indicates the average gas consumption per degree-day for various heat contents. While the fuel consumption in individual cases may vary somewhat from the curve values, these average values are sufficiently accurate for estimating purposes and give very satisfactory results.

The value generally used in the manufactured gas industry for residences is 0.21 cu ft per degree-day per square foot of equivalent steam

radiation (240 Btu) based on the theoretical requirements. A correction for warmer climates is necessary and it is customary to gradually increase the relative fuel consumption below 3,000 degree-days to about 20 per cent more at 1,000 degree-days.

For hot water or warm air heat the fuel consumption is about 0.19 cu ft per degree-day per square foot of equivalent steam *radiation*, that is, per 240 Btu per hour. The actual requirements likewise relatively increase with hot water or warm air systems as the number of degree-days decreases below 3,000. For larger installations, that is, 1,000 sq ft of

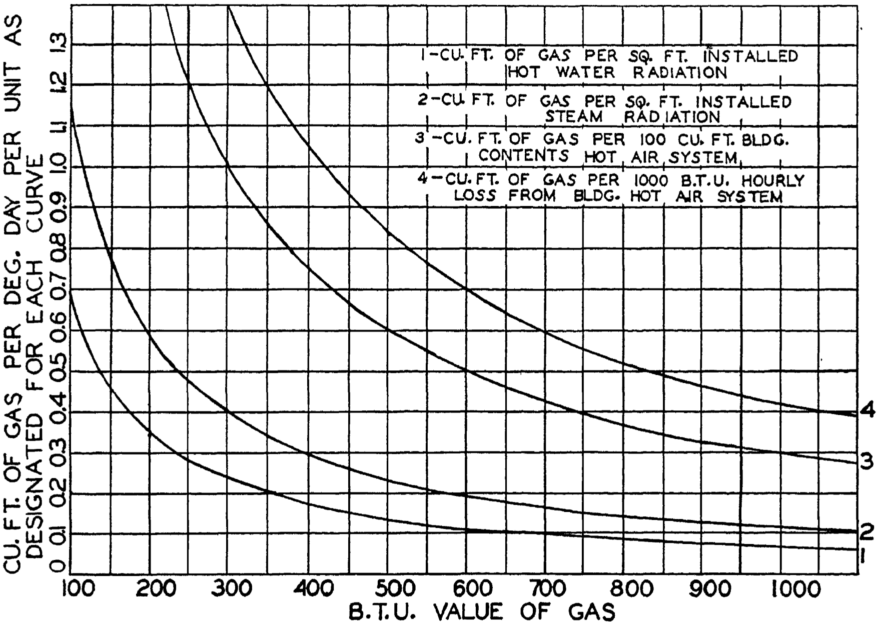


FIG. 2. CHART GIVING GAS REQUIREMENTS PER DEGREE-DAY FOR VARIOUS CALORIFIC VALUES OF GAS AND FOR DIFFERENT HEATING SYSTEMS[a]

[a]This chart is based on an inside temperature of 70 F and an outside temperature of zero. If the *radiation* is installed on the basis of any other temperature difference, multiply the result obtained from this chart by 70, and divide by the actual temperature difference.

theoretical *radiation* and above, there is an increase in efficiency, and a consequent decrease in the fuel consumption per degree-day per square foot of heating surface.

The approximate quantities of steam required in New York City per square foot of heating surface for various classes of buildings are given in Chapter 37.

The preceding discussion on fuel consumption has dealt with the heating requirements of the building irrespective of any air that may be introduced for ventilation purposes other than the normal infiltration of outside air. The heat required for warming air brought into the building for ventilation may be estimated from data given in Chapters 2 and 22.

RELATIVE HEATING COSTS

A comparison of the relative cost of heating with different fuels can be made with even a fair degree of accuracy only when there is a full knowledge of the equipment which will be used with each fuel, and the efficiency with which each will be operated. When proposing to substitute one fuel for another, the yearly cost with the fuel being used can be obtained. The accuracy of the comparison will depend upon the care taken in estimating the cost of the new fuel with the equipment which will be used.

A convenient basis for comparison of various fuels is the cost per million Btu. The formula used in estimating costs for coal is:

$$X = \frac{500 \times c}{C_c \times E_c} \tag{4}$$

where

X = cost of heating with coal in dollars per million Btu.
c = cost of coal in dollars per ton.
C_c = calorific value of coal, Btu per pound.
E_c = over-all or house efficiency for coal, expressed as a decimal.

Example 4. If coal having a calorific value of 13,000 Btu per pound costs \$10.00 per ton, the cost per million Btu, assuming an efficiency of 60 per cent, will be:

$$X = \frac{500 \times 10}{13{,}000 \times 0.60} = \$0.64$$

The formula used in estimating costs for oil is:

$$Y = \frac{1{,}000{,}000 \times p}{C_o \times W \times E_o} \tag{5}$$

where

Y = cost of heating with oil in dollars per million Btu.
p = cost of oil in dollars per gallon.
C_o = calorific value of oil, Btu per pound.
W = weight of oil per gallon, pounds.
E_o = over-all or house efficiency for oil, expressed as a decimal.

Example 5. If oil having a calorific value of 141,000 Btu per gallon ($C_o \times W$) costs 10¢ per gallon, the cost per million Btu, assuming an efficiency of 70 per cent, will be:

$$Y = \frac{1{,}000{,}000 \times 0.10}{141{,}000 \times 0.70} = \$1.01$$

The formula used in estimating costs for gas is:

$$Z = \frac{1000g}{C_g \times E_g} \tag{6}$$

where

Z = cost of heating with gas in dollars per million Btu.
g = average cost of gas, including demand and commodity charges, dollars per thousand cubic feet.
C_g = calorific value of gas, Btu per cubic foot.
E_g = over-all or house efficiency for gas, expressed as a decimal.

Example 6. If manufactured gas, having a calorific value of 535 Btu per cubic foot, costs 60¢ per thousand cubic feet, the cost per million Btu, assuming an efficiency of 80 per cent, will be:

$$Z = \frac{1000 \times 0.60}{535 \times 0.80} = \$1.40$$

PROBLEMS IN PRACTICE

1 ● What two factors are most essential in estimating the fuel consumption for heating a building in a normal season?

The total heat requirements of the building, and the efficiency of combustion.

2 ● What will be the cost per year of heating a building with gas, assuming that the calculated hourly heat loss is 92,000 Btu based on 0 F, which includes 26,000 Btu for infiltration? The design temperatures are 0 F and 72 F. The normal heating season is 210 days, and the average outside temperature during the heating season is 36.4 F. The heating efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 p.m. to 7 a.m. Assume that the price of gas is 7 cents per 100,000 Btu of fuel consumption, and disregard the loss of heat through open windows and doors.

The maximum hourly heat loss will be

$$92{,}000 - \frac{26{,}000}{2} = 79{,}000\ \text{Btu} = H.$$

$$F = \frac{79{,}000 \times (72 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2624.9 \text{ hundred thousand Btu.}$$

The average inside temperature will be

$$\frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The fuel saving will be

$$\frac{100\ (72 - 66.3)}{72 - 36.4} = 16 \text{ per cent.}$$

Hence, the net fuel consumption will be

$$2624.9 - 0.16 \times 2624.9 = 2204.9 \text{ hundred thousand Btu.}$$

$$2204.9 \times 0.07 = \$154.34 = \text{the cost per year of heating the building.}$$

3 ● What factors should be taken into consideration when determining the efficiency at which a fuel will be burned?

Manufacturers' catalogs usually give equipment efficiencies obtained under test conditions. These values do not allow for poor attendance, defects in installation, or poor draft. Such efficiencies do not consider heat radiated from the outside of the equipment, but in many cases this heat is utilized.

4 ● If 20 tons of coal having a calorific value of 13,000 Btu per pound are burned in a warm air furnace and produce 286,000,000 Btu at the bonnet, what is the efficiency of the furnace?

$$\frac{\text{Number of Btu at bonnet}}{\text{Number of tons} \times \text{calorific value} \times \text{number of lb in one ton}} = \text{efficiency.}$$

$$\frac{286{,}000{,}000 \times 100}{20 \times 13{,}000 \times 2000} = 55 \text{ per cent.}$$

5 ● In making degree-day calculations, why is the base of 65 F used for an inside temperature of 70 F?

This base was chosen because data collected from numerous installations show that heat is seldom supplied to a residence when the outdoor temperature is greater than 65 F. It was also found that the amount of fuel consumed varied in almost direct proportion with the difference between 65 F and the outside temperature.

6 ● Make a rough approximation of the amount of coal required to heat a building located in Cleveland, Ohio, assuming that the calculated heating surface requirements are 500 sq ft of steam radiation based on design temperatures of 70 F and 0 F.

Using Fig. 1, the fuel consumption for a design temperature of 0 F is found to be 0.53 tons per thousand degree-days per hundred square feet of heating surface. Cleveland has a heating season equivalent to 6154 degree-days, therefore,

$$0.53 \times 6.154 \times 5 = 16.31 \text{ tons of coal.}$$

7 ● Make a rough approximation of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperatures of 0 F and 70 F. Chicago has 800-Btu mixed gas, and 6315 degree-days.

Using Fig. 2, the fuel consumption for a design temperature of 0 F with 800-Btu gas is found to be 0.08 cu ft of gas per degree-day per square foot of hot water radiation.

$$0.08 \times 6315 \times 1000 = 505{,}200 \text{ cu ft.}$$

8 ● A certain building has a maximum heat loss of 250,000 Btu per hour in −15 F weather. How many tons of fuel will be required to maintain a temperature of 70 F during a 260-day heating season in which the average temperature is 39 F? The heating value of the fuel is 13,200 Btu per pound and the efficiency of combustion is 60 per cent.

$$\frac{250{,}000\ (70 - 39)\ 260 \times 24}{(70 + 15)\ 13{,}200 \times 0.60 \times 2000} = 35.9 \text{ tons.}$$

9 ● Which item may be determined more closely, the heating value of a fuel or the efficiency of its combustion?

The heating values of oil, gas, and solid fuels are closely determinable, whereas the efficiency of burning depends on the particular equipment chosen and the skill used in handling it.

10 ● In an office building, the thermostats are set to maintain 70 F from 7 a.m. to 5 p.m. and 50 F during the rest of the time. When the outside temperature is 30 F, how much saving might be expected because the temperatures are lowered? Under the above conditions the building becomes 50 F by 11 p.m. and warms up to 70 F by 8 a.m.

A temperature of 70 F is maintained during 9 hours, and one of 50 F during 8 hours; the temperature would average about 60 F during the 7 hours required for cooling down and warming up. The average is 60.4 for the 24 hours. (The average temperature calculated would have been 58.3 F, had the warming and cooling periods been neglected.)

The saving is $\left(\frac{70 - 60.4}{70 - 30}\right) \times 100 = \frac{9.6}{4} \times 100 = 24$ per cent.

11 ● How does the heat capacity of a structure influence the saving made by carrying lower temperatures during the night?

The heat storage capacity of the walls prevents rapid dropping of temperatures at night-time and delays the warming up process in the morning. In an extreme case, the building would not reach the lowered temperature by the time the higher temperature is called for in the morning. But under any conditions, the saving made by lowering the temperature can be correctly estimated by using the average temperature observed over the 24-hour period as a factor, as in Question 10.

12 ● What are some of the miscellaneous factors that may cause actual fuel consumption to vary from the theoretical fuel requirements as calculated by the use of heat losses, temperature difference, and fuel burning efficiency?

The opening of windows; abnormally high or low inside temperatures; other sources of heat, such as machinery or lights; sun effect; and unusual winds.

Chapter 30

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator, Enclosed Radiators, Convectors, Code Tests, Gravity-Indirect Heating Systems

THE general terms for heating units are: (1) *radiators*, for direct surfaces, either exposed, enclosed, or shielded; and (2) *convectors*, or concealed heaters, for extended surfaces that are built in as part of an enclosure or cabinet. Some heating units are also available that are a combination of radiators and convectors.

HEAT EMISSION OF RADIATORS AND CONVECTORS

All heating units emit heat by *radiation* and *conduction*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A built-in heating unit in a convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible. The small amount of heat transmitted by radiation to the inside surface of the enclosure diminishes as the surface temperature of the enclosure approaches the surface temperature of the heating unit.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast iron. Catalogs showing the many designs and patterns available now include a junior size which is more compact than the standard unit.

Pipe Coil Radiators

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall type radiators are most frequently used

for this service. When coils are used, the miter type assembly is to be preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is now the *Mb* or 1000 Btu.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot.

Table 1. Variation in Dimensions and Catalog Ratings of 10-Section Tubular Radiators

No. of Tubes	3	4	5	6	7
Width of Radiator, Inches	4.6–5.1	6.0–7.0	8.0–8.9	9.1–10.4	11.4–12.8
Length per Section, Inches	2.5	2.5	2.5	2.5	2.5–3.0
Height with Legs—Inches	Heat Emission—Equivalent Square Feet				
13–14	....	....	....	20	25.0–32.5
16–18	....	....	28.5	....	30.0–38.3
20–21	15.0–17.5	20.0–22.5	25.0–31.2	30	36.7–45.0
22–23	20.0–21.3	25	30.0–33.9	35	40.0–45.2
25–26	20.0–26.7	25.0–27.5	32.5–39.8	37.5–40.0	50.0–53.5
30–32	25.0–30.9	33.3–35.0	40.0–48.6	50	63.3–62.5
36–38	30.0–36.7	40.0–42.5	50.0–56.5	60	70.0–75.4

Output of Tubular Radiators

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

Output of Wall Radiators

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests[1] show that this value will be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

[1]University of Illinois, *Engineering Experiment Station Bulletin No.* 223, p. 30.

Output of Pipe Coils

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. It is customary to use an average emission of 100 Btu per linear foot of 1¼-in. pipe, 10 ft high. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN
Single row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

Effect of Paint

The prime coat of paint on a radiator has little effect on the heat output, but the finishing coat of paint does influence the radiation emission. Since this is a surface effect, there is no noticeable change in the convection loss. Thus, the larger the proportion of direct radiating surface, the greater will be the effect of painting on the radiation. Available tests are on old-style column type radiators which gave results shown in Table 3.

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

RADIATOR No.	FINISH	AREA SQ FT	COEFFICIENT OF HEAT TRANS. BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish	27	1.77	100.5
2	One coat of aluminum bronze	27	1.60	90.8
3	Gray paint dipped	27	1.78	101.1
4	One coat dull black Pecora paint	27	1.76	100.0

[a]Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927).

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[2].

[2]The Heating Effect of Radiators, by Dr. Charles Brabbée (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33).

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2[3]. For the four types of radiators shown, the following conclusions are given:

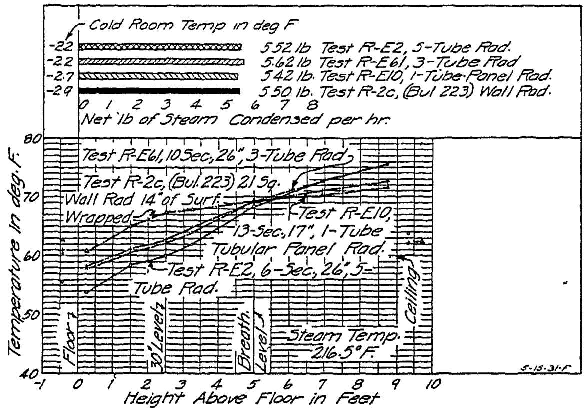


FIG. 1. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 60-IN. LEVEL

Note that the steam condensations are practically the same for all four radiators *when the same air temperature of 69 F is maintained at the 60-in. level.*

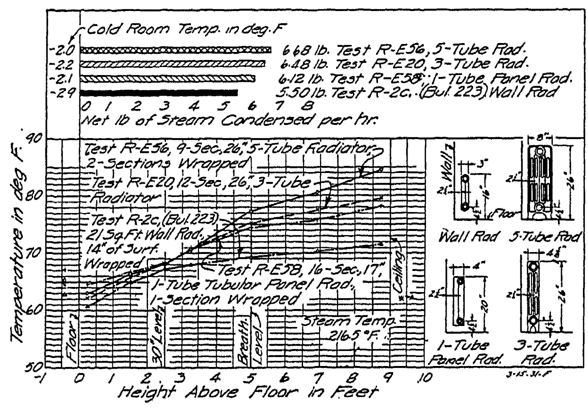


FIG. 2. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 30-IN. LEVEL

Note that the steam condensations are different for all four radiators *when the same air temperature of 68 F is maintained at the 30-in. level.*

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.

[3]Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings[4].

HEATING UP THE RADIATOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old style column type radiators.

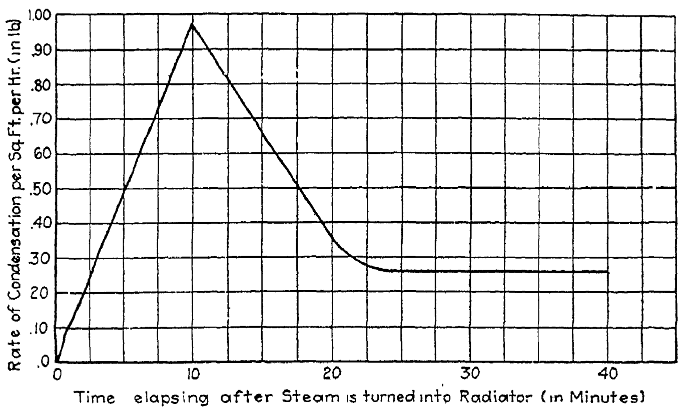


FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Recent investigations[5] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.

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[4] Effect of Two Types of Cast Iron Steam Radiators in Room Heating, by A. C. Willard and M. K. Fahnestock (*Heating, Piping and Air Conditioning*, March, 1930).

[5] University of Illinois *Engineering Experiment Station Bulletins No.* 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

2. The lessened steam consumption may not materially change the radiator heating performance.

3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

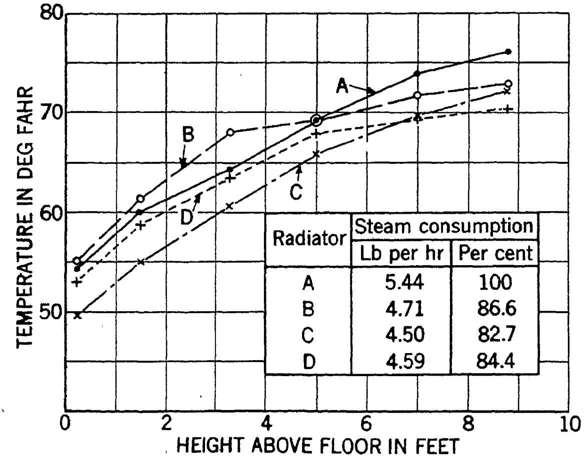


FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

Practically all commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[6] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

CONVECTORS OR CONCEALED HEATERS

Although any standard heating unit (*i.e.*, radiator) may be concealed in a cabinet or other enclosure so that the greatest percentage of heat is

[6]University of Illinois *Engineering Experiment Station Bulletin No.* 230, p. 20.

conveyed to the room by convection, the best results are usually obtained where units of special design are used. Commercially, these specially designed units are built in as part of the enclosing cabinets which are necessary for the proper functioning of these heaters. As distinguished from *radiators*, these gravity convectors have come to be known as concealed heaters. Fig. 5 shows a typical built-in cabinet convector.

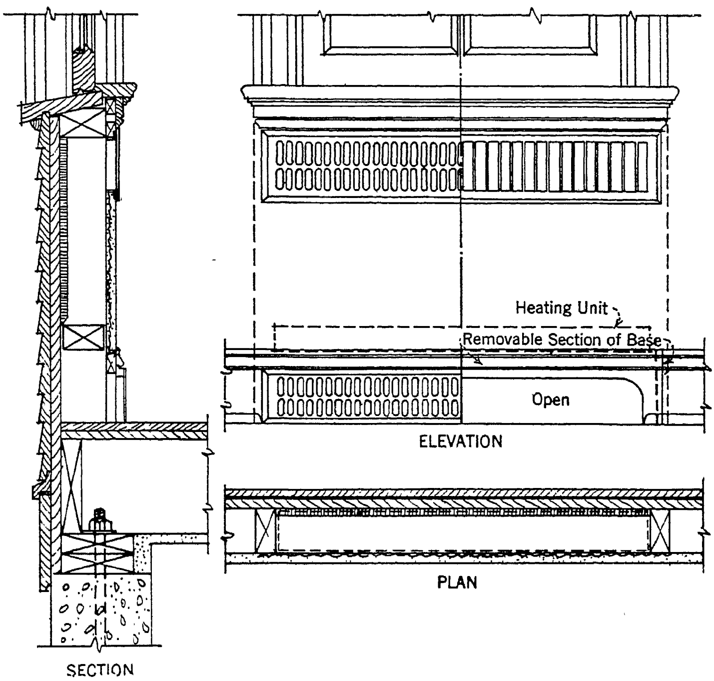


FIG. 5. TYPICAL CONCEALED CONVECTOR USING SPECIALLY DESIGNED HEATING UNIT

The elements or heating units usually consist of a relatively large amount of extended surface which may be integral with the core or assembled over it, making thermal contact by pressure, through solder, or by both pressure and metallic contact. Heating elements may be of cast-iron, cast aluminum, sheet steel, copper, or commercial alloys.

Concealed heaters or convectors maintain room temperatures with low steam consumption due, probably, to their performance characteristics which give reduced air temperatures in the upper level of a room with a directed flow of warm air into the living zone and but little radiant heat to exposed surfaces. *The Concealed Heater Manufacturers Association* has decided to use the A.S.H.V.E. Standard[7] in the formulation of its ratings, but has made a provision that heating effect be included in the ratings in accordance with the following rules:

[7]A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

Convectors 20 in. or lower may have a heating effect of 15 per cent included in their rating; for higher convectors, 1 per cent shall be deducted for each 1 in. increase in height above 20 in. No heating effect shall be included in the ratings for convectors with a cabinet 35 in. from the floor to the top, or higher. All ratings published will show definitely that this heating effect factor is included in the catalog ratings.

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall hanging type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector.

The output of a concealed heater, for any given length and depth, is a variable of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu's. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation.

CODE TESTS FOR RADIATORS AND CONVECTORS

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam, 1932, and Hot Water, 1933).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

Standard test conditions specify either a steam pressure of 1 lb gage (215 F), or hot water at 170 F and a room temperature of 70 F for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \qquad (1)$$

where

H_t = Btu per hour under test conditions.
W_s = condensation in lb per hour.
h_{fg} = latent heat in Btu per lb.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} = \left(\frac{145}{T_s - T_r}\right)^{1.3} \tag{2}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.3} = \left(\frac{150}{T_s - T_i}\right)^{1.3} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s\, H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test, degrees Fahrenheit.
T_r = room temperature during test, degrees Fahrenheit.
T_i = inlet air temperature during test, degrees Fahrenheit.
H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W\,(\theta_1 - \theta_2)\,\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test.
θ_1 = average temperature of inlet water, degrees Fahrenheit.
θ_2 = average temperature of outlet water, degrees Fahrenheit.
t = duration of test, seconds.

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.3} = \left(\frac{105}{\frac{\theta_1 - \theta_2}{2} - T_i}\right)^{1.3} \tag{6}$$

It has been shown that when the exponent 1.3 is used the range of error is less than 5 per cent[8].

GRAVITY-INDIRECT HEATING SYSTEMS[9]

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 6.

[8]Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (*Heating, Piping and Air Conditioning*, August, 1933).

[9]For further information on this subject see A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (edition of 1929) and *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

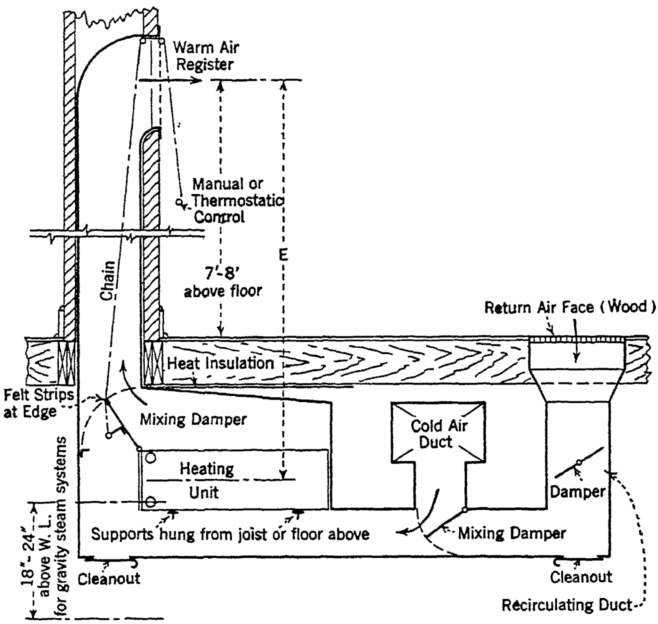


FIG. 6. GRAVITY-INDIRECT HEATING SYSTEM[a]

[a]See *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

The temperature and volume of the air leaving the register must be great enough so that in cooling to room temperature the heat available will just equal the heat loss during the same time. In cases where ventilation is a requirement, the air volume needed may become so large that the entering air temperature will be but slightly above the room temperature. To establish and maintain a constant heat flow, provision must be made for removing the air in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts. As the air flow is maintained by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, especially the runs from the heating units to the base of the vertical warm air flues. Gravity-indirect arrangements, such as illustrated in Fig. 6, are not to be generally recommended for hot water systems unless the water temperature can be maintained at a reasonably high temperature and rapid circulation of the water can be had.

PROBLEMS IN PRACTICE

1 ● What are the principal differences between a radiator and a convector?

A radiator is commonly thought of as a commercial heating unit having a maximum amount of direct heating surface, whereas a convector is a heating device in which the extended or secondary surface may be several times that of the prime surface. The radiator ordinarily has vertical tubular chambers for the heating medium but most convectors have horizontal tubular chambers to which fins are attached so as to form

vertical flues for the passage of air. While radiators are either exposed, enclosed, or shielded, convectors are concealed by means of a tight-fitting enclosure. Radiators are commonly made of cast-iron but convectors may be made of a combination of metals, such as copper and brass, or copper and aluminum.

2 ● How did the term heating effect come into use?

It has been found that a room requiring a radiator of a certain determined capacity could under certain conditions be properly heated, with less temperature gradient between floor and ceiling and with less steam condensation, by the same radiator or by one of a different design having the same commercially rated capacity. This resulted in the use of the term *heating effect* to apply to the useful heat output of a radiator, in the comfort zone of a room, as related to the total input to the radiator.

3 ● What is the effect of enclosing a direct radiator?

This will depend almost entirely on the design of the enclosure. If properly enclosed, a radiator can be made to give better heat distribution below the breathing line and to condense less steam than does an unenclosed radiator giving equal comfort.

4 ● Does paint on a direct radiator affect its heat output?

Aluminum or gold bronze paint tends to reduce the heat output of a direct radiator perhaps 10 per cent, but ordinary non-metallic paint will have little effect on the heat output.

5 ● How can the temperature gradient between the floor and the ceiling of a room be maintained?

Long, low, thin, direct radiators will maintain smaller floor to ceiling temperature differences than high direct radiators. Convectors properly selected and properly installed will accomplish the same result.

6 ● Is the method of enclosing a direct radiator different from that required for a convector?

Generally, yes. An enclosure for a direct radiator should provide a space of at least 2 in. between the radiator and the front and back inside vertical surfaces of the enclosure to utilize the radiant heat to best advantage in heating the air stream passing through the enclosure. The enclosure for a convector should be constructed with as little clearance as possible between the inside vertical surfaces and the convector so as to confine the passage of the air stream through the fins of the convector. An all-over face grille, often used when direct radiators are concealed, should never be used for a convector. The essential requirements for a convector enclosure are an air inlet below the convector, a warm air outlet above it, and sufficient height between the openings to provide a stack effect.

7 ● Is it necessary to make any allowance for the performance of a convector because it is enclosed?

No. The commercial ratings of convectors have been determined by testing the convectors in proper enclosures with grilles in place just as they should be installed for ordinary service.

8 ● On what basis are the capacities of convectors published?

Published ratings of convectors are on the basis of equivalent square feet of direct exposed cast-iron. If any allowance is made for heating effect, the amount of such allowance is generally stated in the manufacturers' catalogs.

9 ● How are fins of convectors attached to the tubes or prime surface?

Tubes or a solid core may be forced through piercings in the fins under pressure, or the

tubes may be expanded into the holes through the fins. In addition a metallic bonding agent is sometimes used to insure permanent contact.

10 ● What is the procedure in selecting a convector when the required amount of radiation is known?

First the limiting factor or factors of the enclosure must be determined so the available size of the wall recess can be found. Manufacturers' catalogs show capacities of convectors of each standard length and depth with varying enclosure heights. From these capacity tables, the proper convector of the required capacity can be selected for the available wall recess. If all three dimensions of the wall recess are insufficient to accommodate a convector of the required capacity, the available height and length can be maintained, but greater depth can be obtained by using a partially recessed enclosure.

11 ● Given a room to be heated to 80 F with outside temperature at zero F. Assume the heat loss under these conditions to be 10,000 Btu per hour. Determine the size of the steam radiator to be installed.

A square foot of radiation is equivalent to a heat emission of 240 Btu per hour under standard conditions of steam at one pound gage pressure (215 F) and surrounding air at 70 F. With surrounding air at 80 F, the heat emission from a radiator will be less. Under these conditions, the heat emission will not be 240 Btu per square foot of catalog rating per hour, but 240 C_s.

$$C_s = \frac{(t_s - t_i)^{1.3}}{(215 - 70)} = \frac{(215 - 80)^{1.3}}{(215 - 70)} = 0.912,$$

and 240 C_s = 240 × 0.912 = 218.5 Btu. Therefore, the size of the radiator to be selected shall have a catalog rating of 10,000 divided by 218.5 or 45.8 sq ft.

Chapter 31

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent System, Gravity Two-Pipe Air-Vent System, One-Pipe Vapor System, Two-Pipe Vapor System, Atmospheric System, Vacuum System, Sub-Atmospheric System, Orifice System, Zone Control, Condensation Return Pumps, Vacuum Pumps, Traps

THE essential features of the common type of steam heating systems are described in this chapter. They may be classified according to the piping arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. Information concerning the design and layout of steam heating systems will be found in Chapter 32.

GRAVITY AND MECHANICAL RETURN

In *gravity systems* the condensate is returned to the boiler by gravity due to the static head of water in the return mains. The elevation of the boiler water line must consequently be sufficiently below the lowest heating units and steam main and dry return mains to permit the return of condensate by gravity. The *water line difference*[1] must be sufficient to overcome the maximum pressure drop in the system and, when radiator and drip traps are used as in two-pipe vapor systems, the operating pressure of the boiler. This applies only to closed circuit systems, where the condensation is returned to the boiler. If the condensation is wasted, no water line difference is required.

In *mechanical systems* the condensate flows to a receiver and is then forced into the boiler against the boiler pressure. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system, but the relative elevation of the boiler water line is unimportant in such cases except that the head on the pump or trap discharge becomes greater as the height of the boiler water line above the trap or pump increases.

There are three general types of mechanical returns in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return pump. Further information on pumps and traps will be presented later in this chapter.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

In the gravity one-pipe air-vent system each radiator has but a single connection through which steam must enter and condensation must

[1]The *water line difference* is the distance between the water line of the boiler and the low point of the water in the dry return main.

return in the opposite direction. Each radiator has an individual air valve.

Up-Feed Gravity One-Pipe Air-Vent System

This system is the most common of all methods of steam heating, due largely to its low cost of installation and its simplicity. As will be seen from Fig. 1, the steam piping rises to a point as high as possible at the boiler and pitches downward from this location until the far end of the main or mains is reached. At the far ends drips are taken off at the low points of the steam mains, are water-sealed below the boiler water line, and then brought back to the boiler in a wet return. Single pipe risers

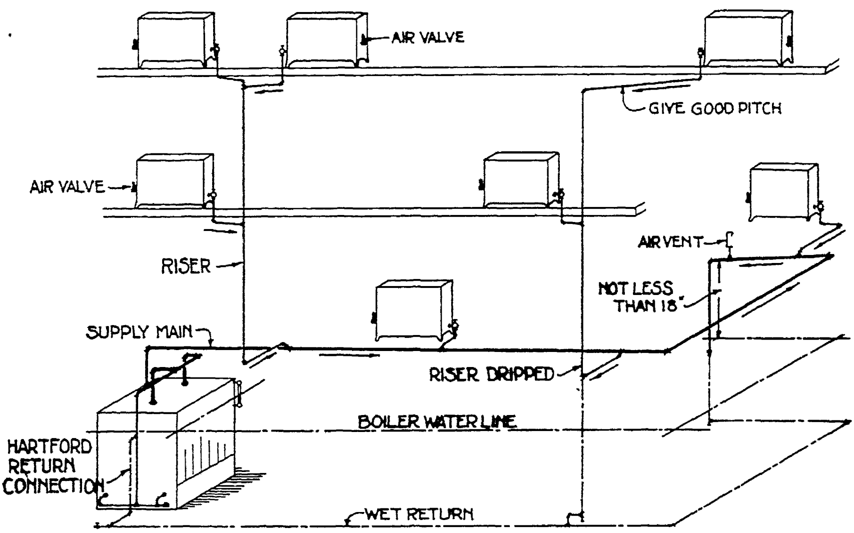


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

are branched off the main or mains to feed the radiators, the steam passing up the riser and the condensation flowing down it. The steam and condensation flow in opposite directions in the riser but after the condensation enters the steam main it flows in the same direction as the steam and is disposed of through the drip connection at the end of the main. In buildings of several stories, it is customary to drip the heel of each riser separately, whereas in one- or two-story buildings this is not necessary. Both types of branches and risers are shown in Fig. 1.

Horizontal branches to radiators and risers should be pitched at least ½ in. in 10 ft downward toward the riser or vertical pipe, and the horizontal branches from the steam main should be graded at least this amount toward the main, except where the heel of the riser is dripped, in which case the branch should pitch down toward the riser drip (Figs. 2 and 3). The return line, if wet, may be run without pitch or may be pitched in either direction, but if it is necessary to carry the return main

overhead for any distance before dropping, the return should slope downward with the flow.

The radiator valves may be of the angle-globe or gate type. They should not be of the straight-globe type because the damming effect of the raised valve seat interferes with the flow of condensation through the valve. Graduated valves cannot be used, as the steam valves on this system must be fully open or closed to prevent the radiators' filling with water. Air valves may be manual or automatic, with or without a check to prevent the re-entrance of expelled air. Usually the automatic type is installed. The greatest source of difficulty with one-pipe steam systems is that the heat is all on or all off, with no intermediate position possible. However, intelligent use of the on-and-off method of manual control gives reasonably satisfactory results.

It is important that the lowest points of the steam mains and heating units be kept sufficiently above the water line of the boiler to prevent

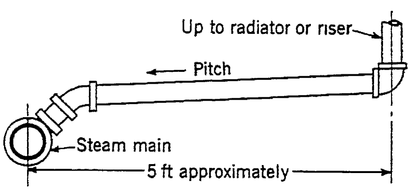


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

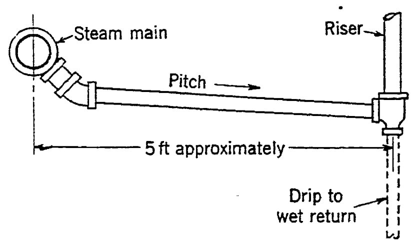


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

flooding, although proper design will eliminate this danger. Usually 18 in. is sufficient but construction limitations frequently make shorter distances necessary. The distance may be checked in the following manner:

Referring to Fig. 4 it will be seen that the water in the wet return is really in an inverted siphon, or U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction of the steam in passing from the boiler to the far end of the main. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise enough farther to produce a flow through the return into the boiler (usually about 3 in. unless the pipes are small or full of sediment), and it will rise still farther if a check valve is installed in the return so as to obtain sufficient head to lift the tongue of the check (usually 4 in. will be necessary).

If a one-pipe steam system is designed, for example, for a total pressure drop of $\frac{1}{8}$ lb, and utilizes an Underwriters Loop[2] instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be $\frac{1}{8}$ of 28 in., or $3\frac{1}{2}$ in. Adding 3 in. to this for the flow through the return main and 6 in. as a factor of safety gives $12\frac{1}{2}$ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of $\frac{1}{2}$ lb, and with a check in the return, would require $\frac{1}{2}$ of 28 in., or 14 in., for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

[2]See discussion of piping details in Chapter 32.

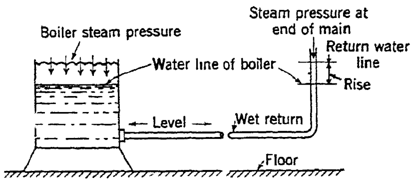


FIG. 4. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

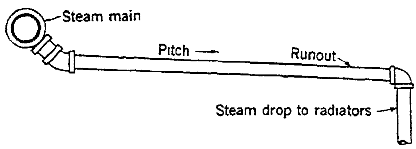


FIG. 6. STEAM RUNOUTS DRIPPING MAIN

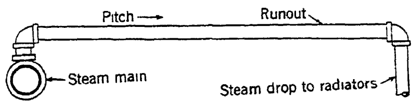


FIG. 7. STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

Down-Feed Gravity One-Pipe Air-Vent System

In the overhead down-feed gravity one-pipe air-vent system there is no change over the *up-feed system* in the radiators, the radiator valves, the air valves, or the radiator runouts as far back as the risers. Beyond this point there are basic differences. The steam is taken from the boiler and carried to the top of the building as near the boiler as possible (Fig. 5). If the run to the main riser is long, or if the riser extends several stories in order to reach the top, the bottom of the riser should be dripped into the wet return. The horizontal main is taken off the top of the riser and grades down from the riser toward all of the drops, each drop taking its share of the main condensation (Fig. 6), or all of the drops except the last may be taken from the top of the main (Fig. 7), the last drop being from the bottom and serving as a drain for the entire main. As the overhead

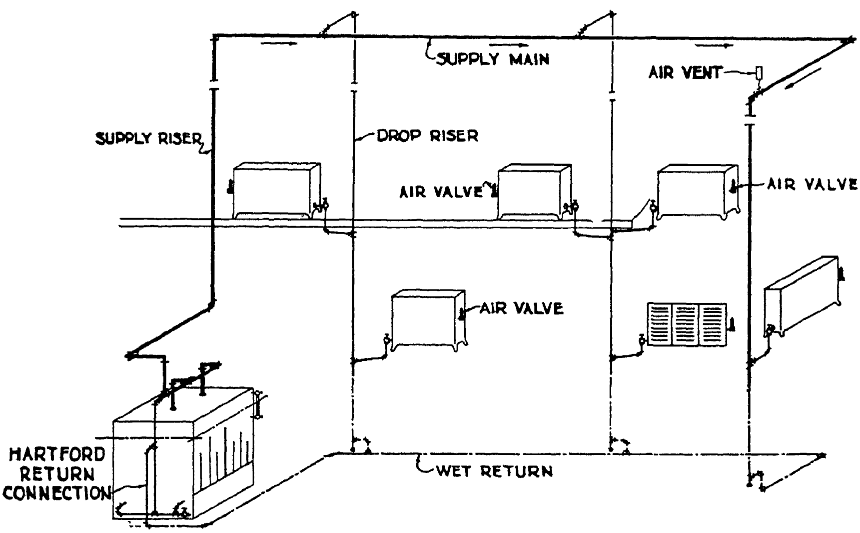


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

main does not carry any condensation from the radiators it is immaterial which method is used. The air vent shown on the main just before the last drop (Fig. 5) may be placed at this point or it may be located at the bottom of the drop under the last radiator connection and sufficiently above the water line of the boiler to prevent flooding.

GRAVITY TWO-PIPE AIR-VENT SYSTEM

The gravity two-pipe system is now considered obsolete although many of these systems are still in use in older buildings. Separate supply and

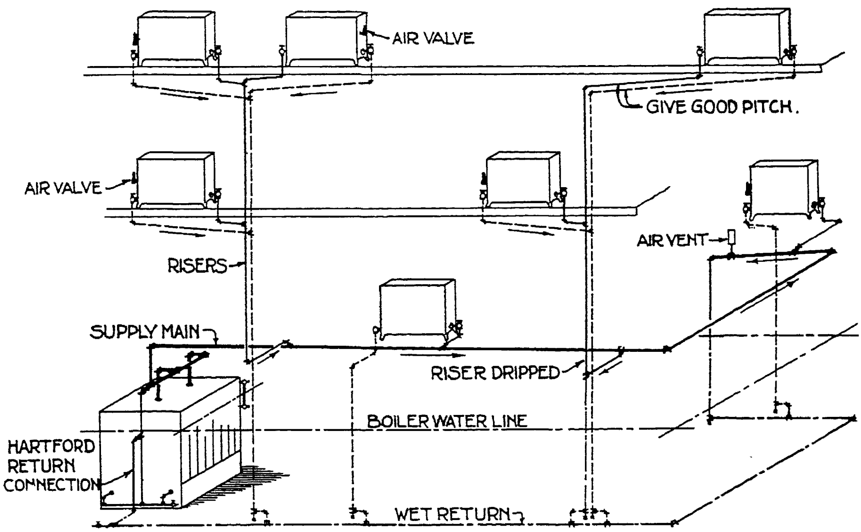


FIG. 8. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

return mains and connections are required for each heating unit; air valves are installed on the heating units and mains; hand valves are installed on the returns.

Up-Feed Gravity Two-Pipe System

This system (Fig. 8) has a steam and a return connection to each radiator. The radiator valves for steam, return, and air are the same as those described for the gravity one-pipe air-vent system. The steam main is run and pitched in the same manner as in the one-pipe system, but the returns from each radiator are connected into a separate return line system which has its risers carried down and joined to a wet return line under the boiler water line level. Where the return has to be kept high to function as a dry return, it is advisable to connect the return risers to the dry return main through water seals about 36 in. deep, as shown in Fig. 9, to prevent steam from one riser entering another and closing the air valves on the nearest radiators.

Down-Feed Gravity Two-Pipe System

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged practically as in the down-feed one-pipe gravity system. The drips at the bottoms of the steam drops and the runouts to the radiators are similar to those shown in Fig. 8 for the up-feed gravity two-pipe system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

ONE-PIPE VAPOR SYSTEM

A vapor system is one which operates under pressures at or near atmospheric and which returns the condensation to the boiler by gravity. The piping arrangement of a one-pipe vapor system is similar to that of

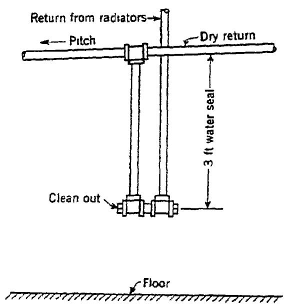


FIG. 9. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

the gravity one-pipe steam system; in fact, one-pipe gravity installations may readily be changed to one-pipe vapor systems by making a few simple alterations. The steam radiator valve is a plug cock which when opened gives a free and unobstructed passageway for water. The automatic air valve is of special design to permit the ready release of air from the radiator and to prevent the return of the air after it is expelled. The air valves on the main are a quick relief type, and the whole system is designed to operate on a few ounces of pressure.

TWO-PIPE VAPOR SYSTEM

Two-pipe vapor systems may be classified as (1) *closed systems* consisting of those which have a device to prevent the return of air after it is once expelled from the system, and which can operate at sub-atmospheric pressures for a period of four to eight hours depending upon the tightness of the system, and (2) *open systems* consisting of those which have the return line constantly open to the atmosphere without a check or other device to prevent the return of air, and which operate at a few ounces above atmospheric pressure.

Under the first classification the essentials are packless graduated valves on the radiators, thermostatic return traps on the returns, and traps on all drips unless they are water sealed. Such a system, illustrated in Fig. 10, should be equipped with an automatic return trap to prevent the water from backing out of the boiler. In this up-feed arrangement the supply piping is carried to a high point directly at the boiler and is graded down toward the end or ends of the supply main, each supply main being dripped at the end into the wet return or carried back to a point near the boiler where it drops down below the boiler water line and becomes a wet return. From this main, runouts are branched off to feed risers or radiators above, these being graded back toward the steam main

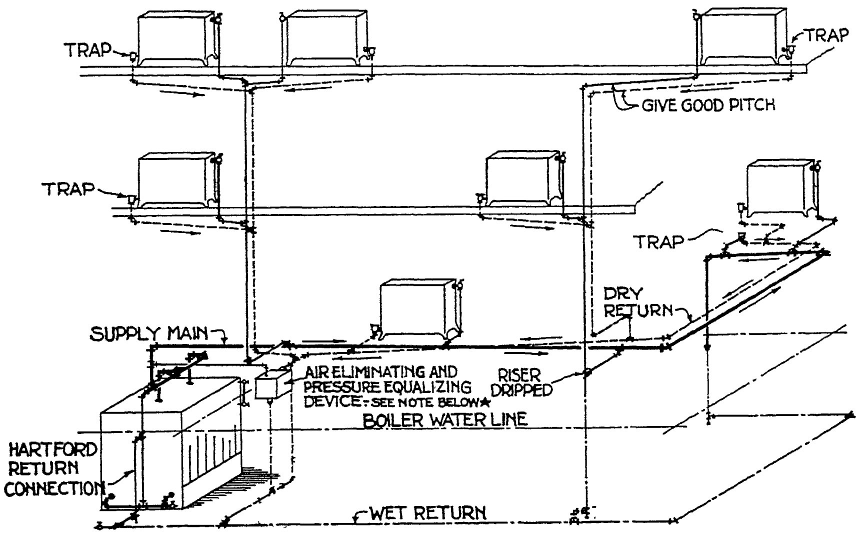


FIG. 10. TYPICAL UP-FEED VAPOR SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

if they are not dripped at the bottom of the riser, or toward the riser if the riser heel is dripped. Both conditions are illustrated in Figs. 2 and 3.

Return risers are connected to each radiator on its return end through thermostatic traps. Their bottoms are connected to the return main through runouts which slope toward the main. The return main itself is sloped back toward the boiler if it is carried overhead; if run wet, the slope may be neglected. An air vent is installed at the point at which the return main drops below the water line. In the simplest cases this vent consists of a ¾-in. pipe with a check valve opening outward, but in certain patented systems special forms of vent valves, designed to allow the air readily to pass out of the system and to prevent its return, are used. A check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which usually is located with the bottom about 18 in. above the boiler water line. Some traps are constructed to permit the bottom's being

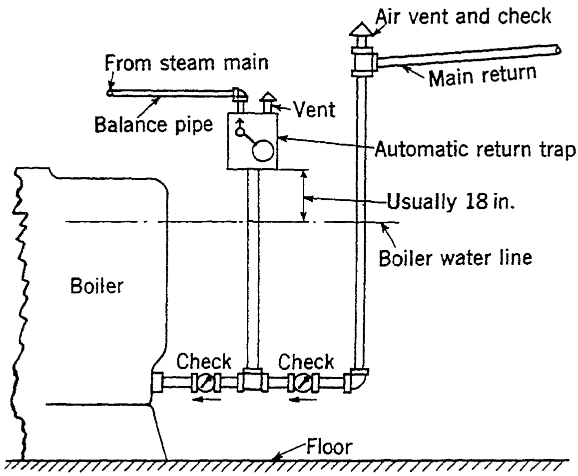


FIG. 11. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

placed as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler (Fig. 11).

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts

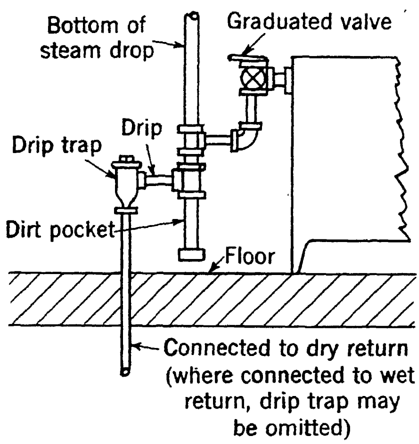


FIG. 12. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

sloped toward the drops (Fig. 6). Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop. Another method of running the steam main, which is not considered as satisfactory but which is practical, is to take the branches off the top of the main (Fig. 7) and to drip the end of the main through the last riser, as illustrated in the down-feed one-pipe system detail shown in Fig. 6. If this is done, the pipe drop at the end or ends of the mains should be enlarged one pipe size to provide capacity for this concentration of the main drip.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of the steam drops should terminate with a dirt pocket above which a drip trap connection is located, as shown in Fig. 12. The returns on a down-feed vapor system are the same as on an up-feed system except that every steam drop must have a drip at the bottom connected either into the return through a trap or into a separate water-sealed drip line below the boiler water line, as illustrated in Fig. 10, in which case the thermostatic traps may be omitted. The runouts to the radiators and the radiator connections of the down-feed system are the same as those of the up-feed system already described.

ATMOSPHERIC SYSTEM

The distinguishing features of the atmospheric system are gravity return to the boiler or to waste, graduated or ordinary radiator valves, no automatic air valves on the radiators, thermostatic traps on the radiator returns, and the venting of all air from the system by means of pipes open to the atmosphere. The returns are open to the atmosphere at all times, usually by extending the return risers to the top of the building where they are either connected together in groups and carried through the roof or extended through the roof individually. Atmospheric systems, either up-feed or down-feed, are often used where the condensation is not returned to the boiler, as in heating systems supplied by high pressure steam through pressure-reducing valves at locations far from the boilers. The returns may be delivered back to the boiler, if desired, by condensation return pumps which are vented to the atmosphere. The return lines in such systems are simply gravity waste lines in which the condensation flows entirely by gravity and is not aided by any pressure difference.

The steam side may be run as that for either up-feed or down-feed two-pipe vapor systems, as the conditions require, and the radiator connections are the same as for vapor systems in that they have graduated valves on the radiator supply ends and thermostatic traps on the radiator return ends. All drips from the supply main and the steam side of the system must pass through thermostatic drip traps before entering the return system where only atmospheric pressure exists. Fig. 13 illustrates a typical scheme of piping used on atmospheric systems.

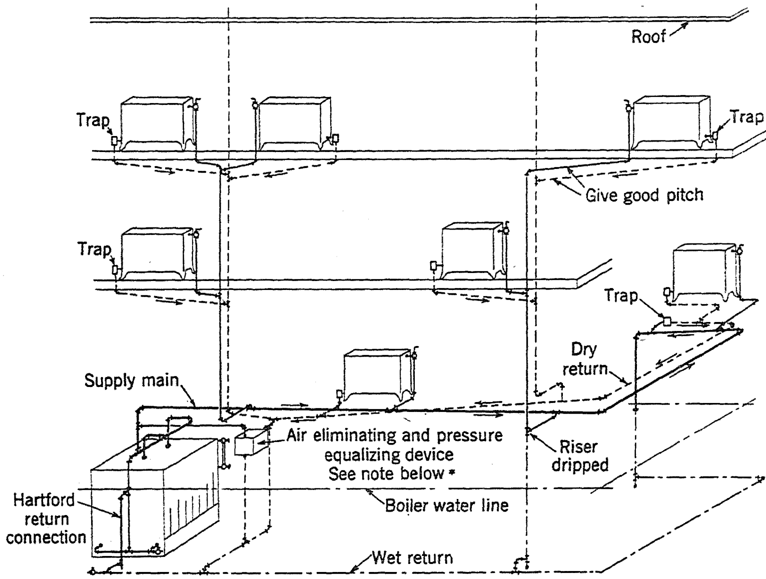


FIG. 13. TYPICAL ATMOSPHERIC SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination

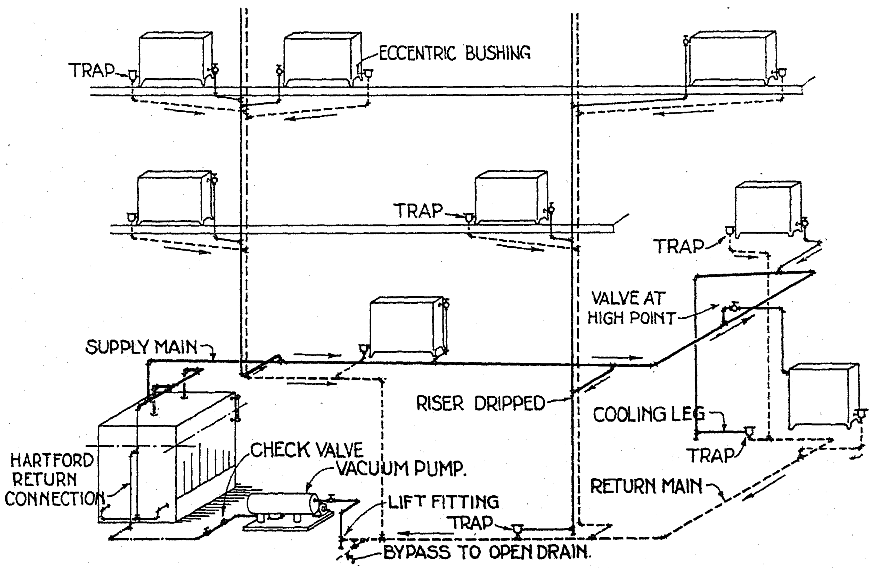


FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

VACUUM SYSTEM

In the vacuum system, a vacuum is maintained in the return line practically at all times but no vacuum is carried on the steam side, and the usual accessories include graduated valves on the radiator supply and thermostatic traps on the radiator return. The air is expelled from the system by a vacuum pump and all drips must pass through thermostatic traps before connecting to the return side of the system.

These systems are often fed from high pressure steam mains through pressure-reducing valves but they may be fed direct from a low-pressure steam heating boiler as shown in Fig. 14, in which a typical up-feed vacuum system is illustrated. The supply main slopes down in the direction of flow; the runouts pitch down toward the riser if the riser is dripped (Fig. 3) or up toward the riser if the riser is not dripped (Fig. 2); both conditions are indicated in Fig. 14. The matter of dripping the risers depends largely on the height of the riser and the judgment of the designer. Ordinarily risers less than three stories high are not dripped and those more than four stories high are dripped, but there is no set rule for this. When risers are dripped the runouts from the steam main may be taken from the bottom if desired and each runout then serves as a drip for the main.

The risers are carried up to the highest radiator connection and are connected to the radiator through runouts sloping back toward the riser. The radiators usually have graduated valves on the supply end, although this is not absolutely necessary. Angle-globe valves and gate valves may be used where graduated manual control is not desirable. The return valves must be of the thermostatic type which will pass air and water but which will close against the passage of steam.

The return risers are carried down to the basement and are connected into a common return line, care being taken that no air pockets exist in the runouts or in the horizontal return main which slopes downward toward the vacuum pump to which it is connected. The air and water are taken by the vacuum pump, which discharges the air from the system and pumps the water back to the boiler, or other receiver, which may be a feed-water tank or a hot well. It is essential on these systems that no connection from the supply side to the return side be made at any point except through a trap.

While the best practice demands a return flowing to the vacuum pump in an interrupted downward slope, in some cases limitations make it necessary to drop the return below the level of the vacuum pump inlet before the pump can be reached. In such event one of the advantages of the vacuum system is that the return can be raised by the suction of the vacuum pump to a considerable height, depending on the amount of vacuum maintained, by means of a lift fitting inserted in the return. When the lift is considerable, several lift fittings are used in steps (Fig. 15), more successful operation being obtained by this method than when the lift is made in one step. If the lift occurs close to the vacuum pump, a special arrangement is used as shown in Fig. 16.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar

runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main (Fig. 6) so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 17). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 18.

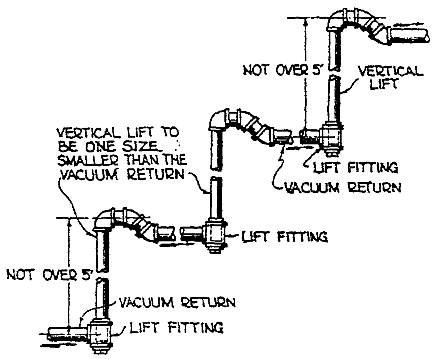


FIG. 15. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

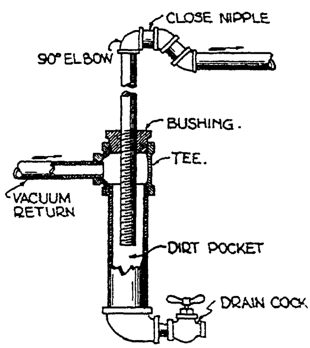


FIG. 16. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

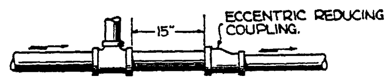


FIG. 17. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

SUB-ATMOSPHERIC SYSTEMS

The sub-atmospheric systems are similar to the vacuum system except that a pump capable of operating up to 25 in. of vacuum is used, and a control is placed on the pump so that the vacuum or absolute pressure carried in the return can be maintained a certain amount below that existing in the steam line to cause a constant circulation. The traps are designed to operate in high vacuum. It is apparent that this system differs from the ordinary vacuum system by having a vacuum on both sides of the system, instead of only on the return side, in order to secure control of the heat emission from the radiators and thus to control the temperature in the building. The system can be operated in the same manner as the ordinary vacuum system when desired.

In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, steam pressure exists in the steam main and

radiators only during the most severe weather, while under average winter temperatures the steam is under a partial vacuum which in mild weather may reach as high as 25 in. This vacuum is largely self-induced by the condensation of the steam in the system when an inadequate supply of steam is being furnished through the control valve which admits it. In the sub-atmospheric system, a control valve is inserted on the steam main of an ordinary vacuum system near the boiler, a high-vacuum pump is substituted for the ordinary type and is supplied with a pressure-difference control, and traps are placed on the radiators and drips which will operate satisfactorily at any pressure from 5 lb gage to 26 in. of vacuum.

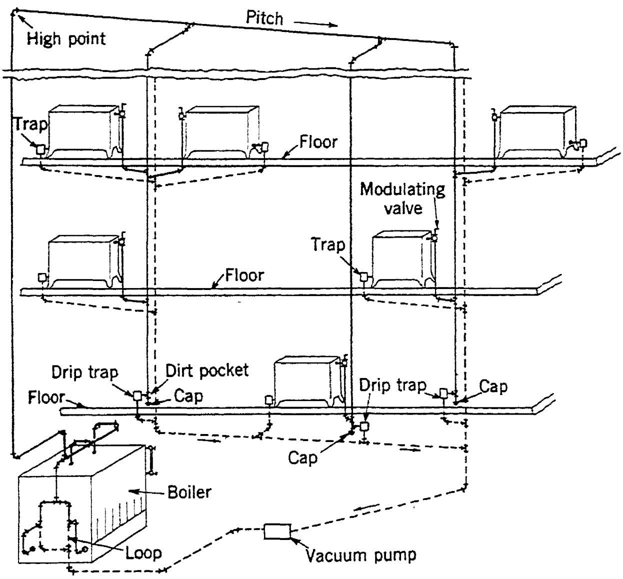


FIG. 18. TYPICAL DOWN-FEED VACUUM SYSTEM

The control valve is a special pressure-reducing valve which may be controlled manually or thermostatically from points selected in the building. The vacuum pump regulator is simply a diaphragm so arranged that, when the vacuum in the return line is insufficient to hold the desired difference in pressure between the steam and return sides of the system, the vacuum pump is automatically started and the vacuum increased to the necessary amount. The actual pressure difference maintained between the two sides of the system is only enough to secure adequate circulation and is often about 2 in. of mercury. This fixed pressure difference between the supply and return sides of the system results in practically constant circulation under all pressure conditions.

In order to distribute the steam equally when the system is being warmed up and also to reduce the amount of steam delivered to the radiators on mild days, orifice plates are used in the graduated radiator control valves. The heat emitted from the radiators in mild weather and

under conditions of high vacuum is not only reduced in proportion to the difference in the steam temperature between that for 2 lb gage and for 25 in. of vacuum but it is reduced still further by a reduction in the amount of steam which can pass through the orifice when the steam is expanded due to the vacuum. This renders possible the control of heat emission from the radiators to a point not indicated entirely by the difference in steam temperatures.

The high-vacuum pumps on this system are equipped with receivers having float control so that the pump can be placed on a receiver-return-pump basis at night if desired so no high vacuum will be carried. One radical difference between this system and the ordinary vacuum system is that no lifts can be made in the return line. The returns must grade downward constantly and uninterruptedly from the radiator return outlet to the inlet on the high-vacuum pump receiver. No attempt should be made to heat service water on this system unless the steam line for water heating is taken off the boiler header back of the heating system control valve, and then only when 2 lb or more will be carried on the boiler at all times.

ORIFICE SYSTEM

Orifice systems of steam heating may have piping arrangements identical with vacuum systems but some of these systems omit both the radiator thermostatic traps and the vacuum pump in cases where the returns are wasted to a sewer or delivered to some type of receiver in which no back pressure exists. The principle on which they operate is embodied in the well-known fact that an orifice will deliver varying velocities when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side no further increase in velocity will be obtained.

As a result, if an orifice is so designed in size as to exactly fill a radiator with steam at 2-lb gage on one side and ¼-lb gage on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 90 \text{ per cent}$$

Should the steam pressure be dropped to ¼ lb gage, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be seen that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, it is simply a question of dropping this main pressure so as to fill any desired portion of the radiator down to the point where the main pressure equals the back pressure in the radiator, at which time no steam will be supplied at all. If orifices throughout a job are designed on a similar basis, all radiators will heat proportionately to the steam pressure within the limits for which the orifices are designed.

Some systems use orifices not only in radiator inlets but also at different points on the main, thus balancing the system to a greater extent. For example, the system may be designed for a particularly long run involving an initial pressure of 3-lb gage on the main and 2 lb at the end of the main,

but each branch from the main may have an orifice for reducing the pressure at it to 2 lb-gage. This is particularly useful for branches near the boiler where the drop in the main has not yet been produced.

Orifice systems using a vacuum pump operate successfully with the ordinary low vacuum type of pump producing 8 to 10 in. of vacuum. They are controlled by various means to regulate the steam pressure. One method is by a thermostat located on the roof to govern the steam pressure by a combination of outside and inside temperatures; another, useful on systems without traps and vacuum pumps, controls the steam pressure manually from temperature indication stations in the building, or automatically by a thermostatically-controlled pressure reduction valve or draft regulator on the boiler; with oil or gas firing, the on-and-off control or a boiler pressure control may be used.

ZONE CONTROL

Certain portions of a building may require more heat at times than others but if the whole building is on one general control, such as would occur with a single piping system with an on-and-off control or with the sub-atmospheric or the orifice systems, it would be necessary to supply sufficient heat to accommodate the coldest portion of the building even though some sections would be overheated. By zoning, each section of a building may be controlled separately.

The sides of the building with different exposures should be considered first, because of the varying effects of the wind and sun. With the prevailing winter winds from the northwest, a simple zoning would place the north and west sides of the building on one system and the south and east sides on another. If the building is large enough to justify the expenditure, a better arrangement would be to place all north walls on one zone, all west walls on a second, all east walls on a third, and all south walls on a fourth.

In case of high buildings, the lowest 8 or 10 stories may be well protected from wind by surrounding buildings, the next 10 stories may have moderate exposure, and above this there may be an unobstructed exposure to gales. On still days the heat demands vertically will vary little, but on windy days there will be a marked difference in the heat requirements for the different horizontal sections. In addition, the *chimney effect* caused by the difference in density between the warm air on the inside of a building and the colder air on the outside will give an air movement which will require zoning to correct. Where such conditions are encountered, the building should be divided horizontally as well as vertically. An arrangement of this character would give 12 zones: namely, north, east, south, and west lower zones; similar middle zones; and similar top zones. Each zone should constitute an individual and separate system of piping with its own supply steam valve (controlled by thermostats in its respective zone) and with its own return or vacuum pump, if one is used. Certain interior areas, such as basements, light well walls and other locations where sun and wind do not affect the conditions, should be placed in still another zone if the most economical results are to be secured.

Zoning has advantages even where individual thermostatic radiator

control is installed whether this be of pneumatic, electric, or the self-contained radiator valve type. By operating the different zones to parallel outside temperature requirements, a large part of the load is taken off the thermostatic controls; they make fewer operations and the radiator follows a more even temperature instead of fluctuating from extreme hot to extreme cold.

CONDENSATION RETURN PUMPS

Condensation return pumps are generally required when the elevation of the boiler with respect to the heating units is such that the condensate will not return by gravity, or when the boiler pressure is greater than that

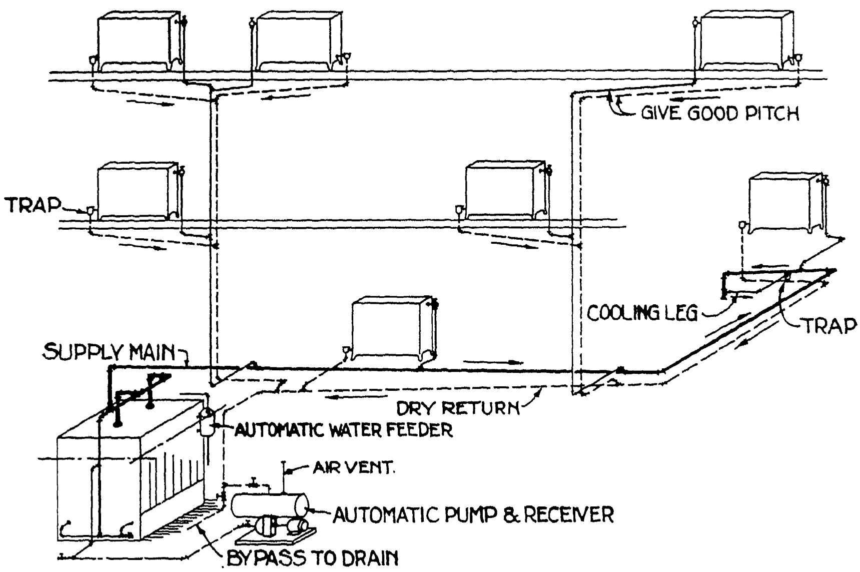


Fig. 19. Typical Installation Using Condensation Pump

supplied the heating units, as in a high-pressure boiler installation supplying steam through a reducing valve to the heating units. The condensate is commonly returned by gravity to a receiver, vented to the atmosphere, from which it flows to the pump.

Condensation return pumps are assembled with tank or receiver and arranged for either continuous operation or for automatic starting and stopping by float control. Any style of water pump may be employed for this service, the power available determining whether the mode of drive shall be steam or electric. The motor-driven, automatic, centrifugal pump and receiver has found wide acceptance in practice for low pressure heating systems.

Fig. 19 shows a typical installation using an automatic condensation return pump and vented receiver. A float control operates the pump whenever sufficient water accumulates. Condensation return pumps are

suitable for use on systems in which the returns are under atmospheric pressure. These include atmospheric systems, orifice systems with open returns, and certain types of vapor systems which operate within a few ounces of atmospheric pressure, but ordinarily do not carry any sub-atmospheric pressure. They may also be used on one-pipe and two-pipe gravity steam systems with a proper arrangement for venting the receiver. In discharging to waste there is no object in using a condensation pump unless the discharge must be elevated.

VACUUM PUMPS

A vacuum heating pump is employed to create a vacuum on the return end of a system to remove air and water and to return the condensate to the boiler or to some other intercepting device that may be employed in plants having mixed systems of heating and other services. Pumps of this classification may be driven by steam or electricity; they may be continuous in operation, or automatic with float or vacuum control in one or more combinations.

For rating purposes[3], vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated under operation at 5½-in. mercury vacuum, and high vacuum pumps are those rated at vacuums above 5½ in.

Return line vacuum pumps are classified in the method of their performance as follows:

a. Those which perform the function of air separation under atmospheric pressure.
b. Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification will handle vacuum steam system condensation coming back by gravity at any temperature up to 205 F without either the sealing or the hurling water flashing into steam. These pumps, to operate under a combined water level and vacuum control, must be equipped with a float-control receiver between the vacuum pump and the system, but where they are intended for continuous operation, they do not require a receiver. Such pumps employ a single vacuum producer which removes the condensate and air from the system and delivers it into a separating chamber under atmospheric pressure from which the condensate is delivered to the boiler or feed water heater. They are constructed on one of the following evacuating and discharge principles:

1. Hydraulic vacuum producer with one pump impeller.
2. Hydraulic vacuum producer with two pump impellers.
3. Water displacement vacuum producer with two pump impellers.
4. Piston displacement vacuum producer with one pump piston.

The second classification of pumps will handle vacuum steam system condensation coming back by gravity at any temperature not exceeding 190 F without the flashing into steam of either the sealing or the hurling water. In order to operate under a combined water-level and vacuum control, these pumps must be equipped with a float-control receiver between the vacuum pump and the system; where intended for continuous operation they do not require a receiver. Such pumps employ a vacuum producing impeller which removes air from the receiver or

[3]See A.S.H.V.E. Standard Code for Testing and Rating return line low vacuum heating pump.

heating system under a partial vacuum and delivers it through an air separator against atmospheric pressure. The condensate is removed from the receiver under a partial vacuum by a separate impeller and is delivered to the boiler or feed water heater. For evacuating and discharge, a water displacement vacuum producer with two pump impellers is used.

Receiver Capacities for Vacuum Pumps

Where receivers are used in connection with vacuum pumps there is a definite relation between the capacity of the receiver and the capacity of the pump. The receiver should have a capacity of not less than 1½ times the volumetric quantity of condensation per minute and should not have such a capacity that the pump will empty the receiver in less than half a minute. Receivers of larger capacities will result in less frequent periods of operation.

Piston Displacement Vacuum Pumps

Piston displacement return-line vacuum heating pumps may be either power or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. While the volumetric displacement for such pumps was formerly figured at 8 to 10 times the volumetric flow of condensation to be handled, the more efficient thermostatic traps used today in connection with vacuum heating systems make it possible to change this proportion so that the volumetric displacement of these pumps may not be less than 6 times the volume of condensation.

Vacuum Pump Controls

In the ordinary vacuum system the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and which cuts out when the vacuum has been increased to the highest point. This is done largely to eliminate the constant starting and stopping of the vacuum pump which would occur if the vacuum were maintained constant. In addition to this control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum in the system. This arrangement makes the vacuum pump primarily a condensation pump and secondarily an air pump.

On the sub-atmospheric systems the high vacuum pump is controlled by a differential regulator which keeps the vacuum in the return line always a few inches higher than that in the steam line and in the radiators.

TRAPS

Traps are used for draining the condensate from radiators, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. The usual functions of a trap are to allow the passage of condensate and to prevent the passage of steam. In addition to these functions, traps are frequently required to allow the passage of air as well as condensate. Traps are also

required to allow the passage of air and to prevent the passage of either water or steam, or both.

In addition, traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may be classified according to the principle of operation as (1) float, (2) bucket, (3) thermostatic, or (4) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used, but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

Thermostatic traps are generally used for draining radiators and heaters, except for very large capacities where bucket, float or blast-type thermostatic traps are used. Thermostatic traps for this service usually

pass both condensate and air and in the case of float and upright bucket traps the air is usually relieved through an auxiliary thermostatic trap in a by-pass around the main trap. Sometimes this auxiliary air trap is an integral part of the trap.

Blast-type thermostatic traps are sometimes used on vacuum heating systems for connecting old one- or two-pipe gravity systems in parallel with vacuum return line systems, in which case the blast-type thermostatic traps should not be provided with auxiliary air by-pass, as the action of this will allow the vacuum to draw air into the old system through its air valves, especially when the steam is wholly or partially

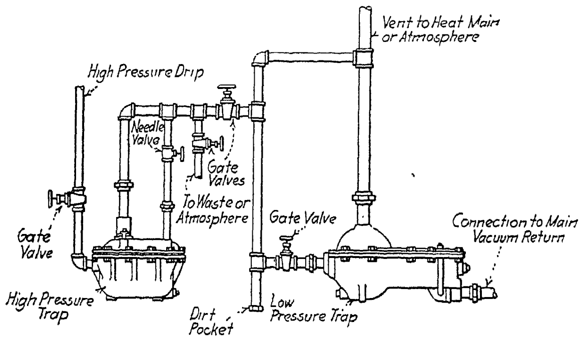


FIG. 20. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW-PRESSURE TRAP

cut off. The air from the returns of such old systems should be relieved just ahead of the traps by means of quick-venting automatic air valves, preferably of the non-return type, especially if the other air valves on the old system are non-return valves.

Tilting traps used for discharging to a higher or a lower pressure are provided with two or three valves operated by the action of the trap. In the case of the two-valve tilting traps, one valve closes a steam inlet and the other valve opens a vent outlet while the trap is filling, and as soon as the trap dumps, the first valve opens the steam inlet and the second valve closes the vent outlet, while the trap discharges. In this type of trap there must be a swinging check-valve on each side of the trap, in addition to the usual by-pass, to prevent the pressure in the trap, while discharging, from backing up through the inlet and the pressure in the discharge line from backing up into the trap while it is filling. This type of trap will blow steam out through the vent while filling, if the pressure on the inlet side is sufficient, and should not be used, therefore, with such pressures unless the vent is properly piped back into the return to a feed water heater, a condenser or a perforated pipe in the bottom of the receiver to which the trap discharges in such a way as to prevent the escape of the steam that comes in with the condensate and passes through the vent. In the three-valve traps of this type there is an extra valve for closing the discharge while the trap is filling.

High pressure traps should not discharge directly into a vacuum return

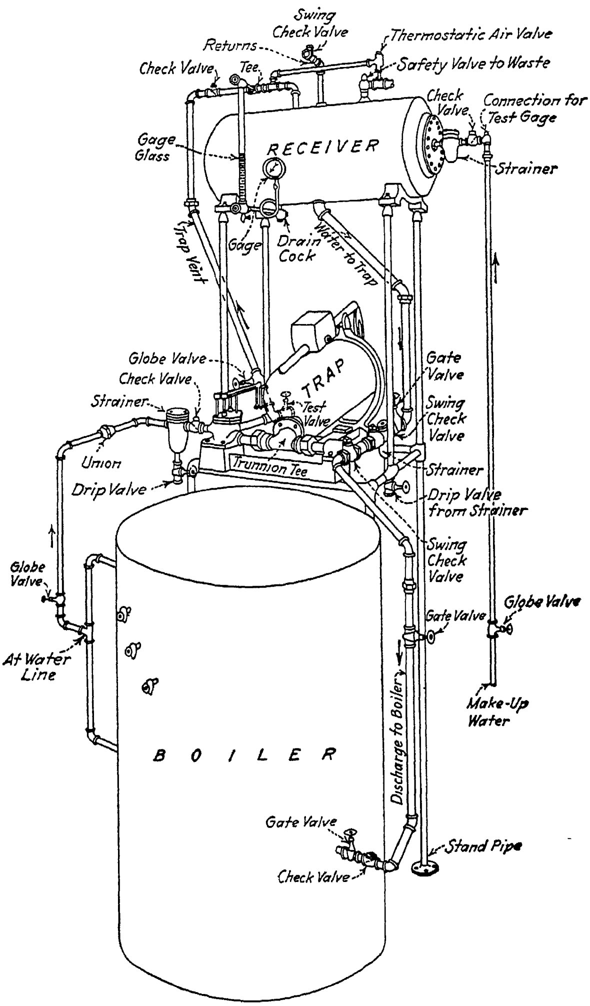


FIG. 21. RETURN TRAP AND RECEIVER FOR AUTOMATIC BOILER FEED

because of the vapor formed by the re-evaporation of a part of the hot condensation. Fig. 20 shows a method which may be used for disposing of the greater part of the vapor of re-evaporation.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation

will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counterbalanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler. Fig. 21 shows a direct return tilting trap and receiver properly connected for automatically feeding a boiler from a system of returns delivering the condensate to the receiver.

PROBLEMS IN PRACTICE

1 ● To what main features does the gravity one-pipe steam system owe its popularity?

To its low cost of installation and to its simplicity.

2 ● How many types of common mechanical returns are there and what are they?

Three: (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum pump.

3 ● In the ordinary vacuum system of steam heating, where does the vacuum usually exist?

On the return side of the system only, between the radiator trap and the vacuum pump. If the radiator supply valve is closed off, the vacuum may extend back through the radiator as far as the supply valve; if an adequate supply of steam is furnished to the system, some vacuum may be developed in the steam main, but neither of these can be termed *normal operation.*

4 ● What is the distinction between the open and the closed vapor systems?

The open vapor system has the return line always open to the atmosphere, while the closed vapor system has an automatic device on the air vent so that air once expelled from the system through the vent cannot re-enter via this route.

5 ● On a vacuum system, what device must be placed on all drips before they enter the vacuum return line?

A thermostatic drip trap or occasionally, where large volumes of condensation are to be handled, a float trap.

6 ● How does the sub-atmospheric system differ in operation from the ordinary vacuum system?

The ordinary vacuum system has pressure in the steam line, and a vacuum produced by the vacuum pump in the return line, usually varying between 5 and 10 in. of water. The sub-atmospheric system may have either a vacuum or pressure on the steam and return

lines, but a constant difference in pressure is maintained between the lines regardless of what pressure or vacuum may be carried. The vacuum, which is generally produced by condensation in the system under conditions of throttled steam supply, may run much higher than in the ordinary vacuum systems.

7 ● What is generally understood by zoning in building steam heating systems?

Zoning is a term applied to the placing of certain sections of a building on a single temperature control instead of having either individual room control or a single temperature control governing the whole building. Zones may be horizontal, such as a single story, a basement, or an attic, or vertical such as the north side, or the west side.

8 ● Why does the water line in the far end of a wet return in a gravity steam system rise higher than the water line in the boiler?

The friction of the steam flowing through the steam main from the boiler to the far end of the system causes a drop in steam pressure at the point where the wet return is connected; consequently, the steam pressure on top of the water in the wet return is less than the steam pressure on top of the water in the boiler, so the water in the end of the wet return rises until a balanced condition is set up.

9 ● On gravity one-pipe systems as indicated in Fig. 1 and Fig. 3, why is the drip on the steam runout connected to wet return?

Because if it were connected to dry return, the pressure drops to two different points would not necessarily be the same and the system would short circuit.

10 ● Why cannot graduated valves be used on a one-pipe system?

Partial opening of valves would restrict flow to such an extent that the radiator could not drain properly and would fill with water.

11 ● What advantage is there to an air valve with a check to prevent the reentrance of expelled air?

A system equipped with such valves builds up a vacuum and holds the heat longer. With proper controls on the boiler, lower radiator temperatures can be maintained in mild weather, giving better plant efficiency.

12 ● With a one-pipe steam heating plant designed for a total pressure drop of ¼ lb with a check valve on the return, how high must the lowest part of the steam main be above the boiler water line?

Water line difference (¼ × 28)	7 in.
Flow head required	3 in.
Friction head of check valve	4 in.
Factor of safety	6 in.
Total required	20 in.

13 ● What are the essentials of a two-pipe closed vapor system?

Packless graduated valves on radiators; thermostatic return traps on returns and drips; an automatic return trap to prevent water from backing out of the boiler.

14 ● Why must the automatic return trap on two-pipe vapor systems be about 18 in. above the boiler water line?

That height is necessary to overcome water line difference owing to pressure drop and friction in pipe and fittings.

15 ● What is the difference between the systems illustrated in Fig. 10 and Fig. 13?

The risers and the air eliminator in Fig. 13 are vented to atmosphere.

16 ● What is the difference between a vacuum pump and a condensation return pump?

The vacuum pump produces and maintains a vacuum in the return lines whereas the condensation return pump returns the condensation back to the boiler. The relation of the boiler to the heating units is such that the condensate will not return by gravity.

17 ● What is the function of a trap?

The usual function is to allow the passage of condensate and air and to prevent the passage of steam.

18 ● Under what conditions is it advisable to use a combination float and thermostatic trap?

Where unusual capacities are required, as on large mains or blast coils.

19 ● Why should the discharge from high pressure traps not go directly into a vacuum return main?

Because of its higher temperature, the high pressure condensate would immediately flash into steam in the vacuum main and cause difficulty with the vacuum pump.

20 ● What is the function of the automatic return trap?

To insure the return of condensate to the boiler when the operating condition is such that the boiler pressure exceeds the static head on the returns.

Chapter 32

PIPING FOR STEAM HEATING SYSTEMS

Flow of Steam in Pipes, Pipe Sizes, Tables for Pipe Sizing, Sizing One-Pipe Gravity Air Vent Systems, Two-Pipe Gravity Air Vent Systems, Two-Pipe Vapor Systems, Atmospheric Systems, Vacuum Systems, Sub-Atmospheric Systems, Orifice Systems, High Pressure Steam, Expansion in Steam and Return Lines, Piping Connections and Details, Boiler Connections, Hartford Return Connection

THE design of a steam heating system should be considered under four headings, namely, (1) the details of the heating units, (2) the arrangement of the general piping scheme, (3) the details of connections, and (4) the sizing of the lines. Items 1 and 2 are covered in Chapters 30 and 31, respectively, while this chapter considers the two latter items.

The functions of piping are to supply the heating units with steam and to remove the condensation. In some systems both the air and condensation are removed from the heating units by the return piping. To accomplish this effectively, the distribution of the steam should be efficient and equitable, without noise, and the returns should be as short as possible. When air is handled its escape should be facilitated to the utmost since an air-bound system will not heat properly. Condensation takes place in a steam system not only in the heating units, but throughout the piping system as well, and the returns also condense any steam or vapor that may be contained. At the same time part of the condensation may flash back into steam when the vacuum or pressure in the return is considerably below the steam pressure.

It is essential that steam piping systems not only distribute steam at full load but also at partial loads, as the average winter demand is less than half of the demand in most severe outside temperatures. Furthermore, in heating up rapidly the load on the steam main may exceed the maximum operating load even in extreme weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature. This may require more heat than would be emitted from the system itself after it once is thoroughly heated.

STEAM FLOW

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam, and the pressure drop through the pipe. This relationship has been established by Babcock in the following formula:

$$P = 0.0000000367\left(1 + \frac{3.6}{d}\right)\frac{W^2L}{Dd^5} \qquad (1)$$

or

$$W = 5220\sqrt{\frac{PDd^5}{\left(1 + \frac{3.6}{d}\right)L}} \qquad (2)$$

where

P = loss in pressure, pounds per square inch.
d = inside diameter of pipe, inches.
L = length of pipe, feet.
D = weight of 1 cu ft of steam.
W = weight of steam flowing per hour, pounds.

Example 1. How much steam will flow per hour through 100 ft of 2-in. pipe if the initial pressure is 1.3 lb per square inch and the pressure drop is 1 oz?

Solution. $P = \frac{1}{16}$ = 0.0625 lb; d = 2.067 in. (Table 1, Chapter 34); L = 100 ft; D = 0.04038 lb (Table 7, Chapter 1). Substituting these values in Formula 2:

$$W = 5220\sqrt{\frac{0.0625 \times 0.04038 \times 2.067^5}{\left(1 + \frac{3.6}{2.067}\right)100}} = 97.2 \text{ lb per hour.}$$

Formula 2 does not allow for entrained water in low-pressure steam, condensation in pipe, and roughness in commercial pipe as found in practice.

The latent heat of steam (h_{fg}) at atmospheric pressure (Table 7, Chapter 1) is 970.2 Btu per pound. Inasmuch as the heat emission of an equivalent square foot of heating surface (radiation) is 240 Btu, 1 lb of steam at this pressure will supply $\frac{970.2}{240}$ or 4.04 sq ft of equivalent heating surface. This figure is usually taken as 4 even. In Example 1, the weight of steam flowing per hour would therefore supply 4 × 97.2 or 388.8 sq ft of equivalent heating surface.

PIPE SIZES

The determination of pipe sizes for steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. Unusual conditions in the building to be heated.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems

TABLE 1. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0.68	45	10,961	11.3
1¼	17.6	0.66	98	23,765	24.5
1½	20.0	0.66	152	36,860	38.0
2	23.0	0.57	288	69,840	72.0
2½	26.0	0.54	464	112,520	116.0
3	29.0	0.48	799	193,600	199.8
3½	31.0	0.44	1144	277,000	286.0
4	32.0	0.39	1520	368,000	380.0

INSTRUCTIONS FOR USING TABLE 1

1. Capacities given in Table 1 should never be exceeded on one-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5).

specially designed for varying initial pressures, such as the sub-atmospheric, the orifice, and the vapor systems which normally operate under partial vacuums; (4) there is sufficient difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, and the dry return, when considered in relation to the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first,* the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second,* a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 1 and 2 for vertical risers and in Table 3 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the

TABLE 2. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	-----	40	9550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 2

1. The capacities given in this table should never be exceeded on two-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5.)

quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, and (3) the quantity of condensate flowing against the steam.

Two factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1-in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 4). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 5). All of the capacity tables given in this chapter include a factor of safety. However, the pipe on which Table 4 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

TABLE 3. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES[a]

Pitch of Pipe in Inches per 10 Ft

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from A.S.H.V.E. Research Laboratory.

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 6 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLE 4. PER CENT DIFFERENCE IN CAPACITY FOR CARRYING STEAM AND CONDENSATE DUE TO VARIATION OF PIPE SIZE AND SMOOTHNESS[a]

	MAXIMUM CONDENSATION, LB PER HOUR			
Size of pipe	¾ In.	1 In.	1¼ In.	1½ In.
Minimum	14.00	24.89	45.42	70.50
Maximum	15.20	30.08	52.08	82.00
Per cent variation	8.6	20.8	14.7	16.3

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 5. EFFECT OF REAMING ENTRANCE TO ONE-INCH ONE-PIPE RISERS[a]

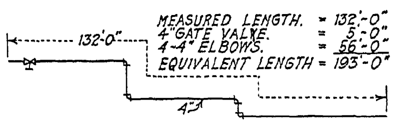
	MAXIMUM CAPACITY OF RISER	PER CENT DECREASE
Reamed entrances	24.7 lb per hour	0.0
Rounded entrances	23.9 lb per hour	3.2
Squared entrances	22.2 lb per hour	10.1
Three wheel cutter	19.2 lb per hour	22.2
Single wheel cutter	17.6 lb per hour	28.7

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

Table 6. Length in Feet of Pipe to be Added to Actual Length of Run—Owing to Fittings—to Obtain Equivalent Length

Size of Pipe Inches	St'd. Elbow	Side Outlet Tee	Gate Valve	Globe Valve	Angle Valve
	Length in Feet to be Added to Run				
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING[1]

Factors determining the size of a steam pipe and its allowable limit of capacity are as follows:

1. Pipe condensate flowing with steam.
2. Pipe condensate flowing against steam.
3. Pipe and radiator condensate flowing with steam.
4. Pipe and radiator condensate flowing against steam.

It is apparent that (3) and (4) are practically limited to one-pipe systems while (1) and (2) cover all other systems.

Tables 7 and 8, worked out for determining pipe sizes, have their columns lettered continuously, Columns *A* through *L* being in Table 7, and *M* through *EE* in Table 8. In the following text, reference made to columns will be by letter. The tables are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation[2] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

[1]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of The Guide has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[2]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of The Guide to gradually eliminate the empirical expression *square foot of equivalent direct radiation*, *EDR*, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A.S.H.V.E.

TABLE 7. STEAM PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

	CAPACITIES OF STEAM MAINS AND RISERS								SPECIAL CAPACITIES FOR ONE-PIPE SYSTEMS ONLY		
	DIRECTION OF CONDENSATION FLOW IN PIPE LINE										
PIPE SIZE IN.	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only		Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
	1/32 lb or ½ Oz Drop	1/24 lb or ⅔ Oz Drop	1/16 lb or 1 Oz Drop	⅛ lb or 2 Oz Drop	¼ lb or 4 Oz Drop	½ lb or 8 Oz Drop	Vertical	Horizontal			
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
¾	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	....	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	....	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	....	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	....	1,110
5	3,214	3,712	4,546	6,429	9,092	12,858	....	3,150	....	....	2,180
6	5,276	6,094	7,462	10,553	14,924	21,105	....	....	....	....	....
8	10,983	12,682	15,533	21,967	31,066	43,934	....	....	....	....	....
10	20,043	23,144	28,345	40,085	56,689	80,171	....	....	....	....	....
12	32,168	37,145	45,492	64,336	90,985	128,672	....	....	....	....	....
16	60,506	69,671	84,849	121,012	169,698	242,024	....	....	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 7.

Table 7 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve, and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 8 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 7. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

TABLE 8. RETURN PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter M through EE)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

CAPACITY OF RETURN MAINS AND RISERS

Pipe Size Inches	1/32 Lb or ½ Oz Drop per 100 Ft			1/24 Lb or ⅔ Oz Drop per 100 Ft			1/16 Lb or 1 Oz Drop per 100 Ft			⅛ Lb or 2 Oz Drop per 100 Ft			¼ Lb or 4 Oz Drop per 100 Ft			½ Lb or 8 Oz Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
Mains																		
¾	....	....	....	....	....	326	....	....	400	....	....	568	....	....	800	....	....	1,130
1	500	248	....	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	....	....	1,977
1¼	850	520	....	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	....	....	3,390
1½	1,350	822	....	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	....	....	5,370
2	2,800	1,880	....	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	....	....	11,300
2½	4,700	3,040	....	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	....	....	18,925
3	7,500	5,840	....	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	....	....	30,230
3½	11,000	7,880	....	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	....	....	45,200
4	15,500	11,700	....	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	....	....	62,180
5	....	....	....	....	....	31,500	....	....	38,700	....	....	54,920	....	....	77,400	....	....	109,300
6	....	....	....	....	....	50,450	....	....	62,000	....	....	88,000	....	....	124,000	....	....	175,100
Risers																		
¾	....	190	....	....	190	570	....	190	700	....	190	994	....	190	1,400	....	....	1,977
1	....	450	....	....	450	976	....	450	1,200	....	450	1,704	....	450	2,400	....	....	3,390
1¼	....	990	....	....	990	1,547	....	990	1,900	....	990	2,696	....	990	3,800	....	....	5,370
1½	....	1,500	....	....	1,500	3,256	....	1,500	4,000	....	1,500	5,680	....	1,500	8,000	....	....	11,300
2	....	3,000	....	....	3,000	5,453	....	3,000	6,700	....	3,000	9,510	....	3,000	13,400	....	....	18,925
2½	....	....	....	....	....	8,710	....	....	10,700	....	....	15,910	....	....	21,400	....	....	30,230
3	....	....	....	....	....	13,020	....	....	16,000	....	....	22,710	....	....	32,000	....	....	45,200
3½	....	....	....	....	....	17,910	....	....	22,000	....	....	31,220	....	....	44,000	....	....	62,180
4	....	....	....	....	....	31,500	....	....	38,700	....	....	54,920	....	....	77,400	....	....	109,300
5	....	....	....	....	....	50,450	....	....	62,000	....	....	88,000	....	....	124,000	....	....	175,100

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2-lb gage?

Solution. It will be assumed, if the measured lengtn of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ lb, while if the total drop were $\frac{1}{2}$ lb, the drop per 100 ft would be $\frac{1}{20}$ lb. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

One-pipe gravity air-vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection*, use Column *K*.

6. For the *dry return main*, use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{24}$-lb drop is being used, the steam main and dripped runouts would be sized from Column *C;* radiator runouts and undripped riser runouts from Column *L;* up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{24}$ lb); the dry return from Column *R;* and the wet return from Column *Q*.

With a $\frac{1}{32}$-lb drop the sizing would be the same as for $\frac{1}{24}$ lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Example 3. Size the one-pipe gravity steam system shown in Fig. 1 assuming that this is all there is to the system or that the riser and run shown involve the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ lb the drop per 100 ft will be slightly less than $\frac{1}{16}$ lb. It would be well in this case to use $\frac{1}{24}$ lb, and this would result in the theoretical sizes indicated in Table 9. These theo-

TABLE 9. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 1

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED (SQ FT)	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to radiators	....	100	2	2
Branches to radiators	....	50	$1\frac{1}{4}$	$1\frac{1}{4}$
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	$2\frac{1}{2}$	$2\frac{1}{2}$
Riser	*c* to *d*	400	$2\frac{1}{2}$	$2\frac{1}{2}$
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to riser	*f* to *g*	600	$3\frac{1}{2}$	$3\frac{1}{2}$
Supply main	*g* to *h*	600	3	3
Branch to supply main	*h* to *j*	600	$2\frac{1}{2}$	3
Dry return main	*f* to *k*	600	$1\frac{1}{4}$	2
Wet return main	*k* to *m*	600	1	2
Wet return main	*m* to *n*	600	1	2
Wet return main	*n* to *p*	600	1	2

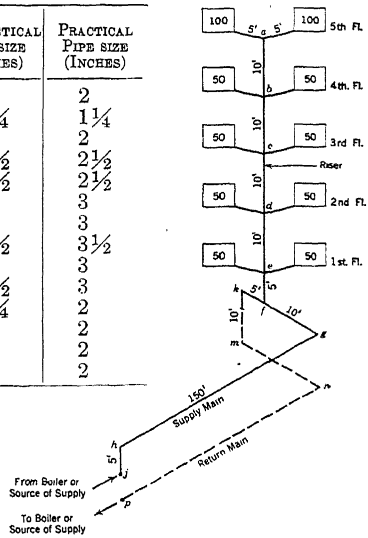


FIG. 1. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

retical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

Notes on Gravity One-Pipe Air-Vent Systems

1. Radiator runouts over 8 ft long should be increased one pipe size.
2. Pitch of mains should be not less than ⅛ in. in 10 ft.
3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft.
4. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should be not less than half its diameter at its largest part.
5. Supply mains, branches to risers, or risers, should be dripped where necessary.

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of ½ lb the drop per 100 ft would be $\frac{\frac{1}{2}}{8}$ or $\frac{1}{16}$ lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of ¼ lb is desired, the drop per 100 ft would be $\frac{1}{32}$ lb

and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be ⅛ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is $\frac{1}{64}$ lb or $\frac{1}{32}$ lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 8 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ lb to ¼ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ lb divided by 4, or $\frac{1}{32}$ lb. In this case the steam mains would be sized from Column *B*; the radiator and undripped riser runouts from Column *I*; the risers from Column *B*,

because Column H gives a drop in excess of $\frac{1}{32}$ lb. On a down-feed system, Column B would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over $\frac{1}{32}$ lb. The return risers would be sized from the lower portion of Column O and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column N. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Radiator runouts over 8 ft long should be increased one pipe size.
2. Pitch of mains should be not less than $\frac{1}{8}$ in. in 10 ft.
3. Pitch of horizontal runouts to risers and radiators should be not less than $\frac{1}{2}$ in. in 10 ft.
4. In general it is not desirable to have a supply main smaller than 2 in., and when the supply main is 3 in. or over at the boiler or pressure reducing valve it should be not less than $2\frac{1}{2}$ in. at the far end.
5. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return. The drip for a vapor system may be connected into the dry return through a thermostatic drip trap.

VACUUM SYSTEMS

Vacuum systems are usually employed in large installations and have total drops varying from $\frac{1}{4}$ to $\frac{1}{2}$ lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of $\frac{1}{2}$ lb divided by 12, or $\frac{1}{24}$ lb. In this case the steam main would be sized from Column C, and the risers also from Column C (Column H could be used as far as critical velocity is concerned but the drop would exceed the limit of $\frac{1}{24}$ lb). Riser runouts, if dripped, would use Column C but if undripped would use Column I; radiator runouts, Column I; return risers, lower part of Column S; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed $\frac{1}{8}$-lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.
2. Radiator runouts over 8 ft long should be increased one pipe size.
3. Pitch of mains should be not less than $\frac{1}{8}$ in. in 10 ft.
4. Pitch of horizontal runouts to risers and radiators should be not less than $\frac{1}{2}$ in. in 10 ft.
5. In general it is not considered desirable to have a supply main smaller than 2 in. When the supply main is 3 in. or over, at the boiler or pressure reducing valve, it should be not less than $2\frac{1}{2}$ in. at the far end.
6. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a thermostatic trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a thermostatic trap to prevent the steam from entering the return line.
7. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 31 under *Up-Feed Vacuum Systems*.

ATMOSPHERIC SYSTEMS

The sizing of the supply and return piping on atmospheric systems is practically identical with the sizing used for vacuum systems and the same notes apply, except that no lift can be made in the return line.

SUB-ATMOSPHERIC SYSTEMS

Any properly pitched, correctly sized vacuum system without a lift may be used as a sub-atmospheric system when the proper equipment is substituted for the ordinary vacuum pump, traps, and controls. On new systems manufacturers usually recommend a drop on the steam line of between ¼ and ⅜ lb for the total run, and suggest adding 25 ft to the total equivalent length of run to insure that the steam gets through to the last radiator.

The same notes apply to these systems as for vacuum systems, except that no lifts can be made in the returns.

ORIFICE SYSTEMS

The orifice systems can be operated with any piping system suitable for vacuum operation, according to experienced designers. Because these systems vary considerably in detail, it is advisable to consult the manufacturer of the particular system contemplated for recommendations.

The same notes apply to these systems as to vacuum systems, except that lifts cannot be made in the returns of orifice systems if a vacuum pump is used.

HIGH PRESSURE STEAM

When steam heating systems are supplied with steam from a high pressure plant, one or more pressure-reducing valves are used to bring the pressure down to that required by the heating system. It has been considered good practice to make the pressure reductions in steps not to exceed 50 lb in each case. For example, in reducing from 100-lb gage to 2-lb gage, two pressure reducing valves would be used, the first reducing the pressure from 100-lb gage to 50 lb and the second reducing the pressure from 50-lb gage to 2-lb gage. Valves are available that will reduce 100 lb in one step, and it is questionable whether two valves are now required for initial pressures of 150 lb or less.

The pressure-reducing valve, or pressure-regulator as it is sometimes termed, has ratings which vary 200 to 400 per cent. Some of these ratings are based on arbitrary steam velocities through the valve of 5,000 to 10,000 fpm and it is assumed that the valve when wide open has the same area as the pipe on the inlet opening of the valve. It is well known that steam flowing through an orifice increases its velocity until the pressure on the outlet side is reduced to 58 per cent of the absolute pressure on the inlet side, and that with further reduction of pressure on the outlet side little change in velocity will be obtained. As practically all pressure-reducing valves used for steam heating work lower the steam pressure to less than 58 per cent of the inlet pressures, only the maximum velocity through such valves need be considered. If it is assumed that the valve, when fully open, has an area equal to that of the inlet pipe size,

TABLE 10. CAPACITIES OF PRESSURE-REDUCING VALVES
(100-LB GAGE DOWN TO ANY PRESSURE—52 LB OR LESS)

INLET NOMINAL PIPE DIAMETER (INCHES)	POUNDS STEAM PER HOUR AT 100-LB GAGE	EQUIVALENT DIRECT RADIATION SQ FT AT ¼ LB	EQUIVALENT DIRECT RADIATION SQ FT AT 1/3 LB
½	866	3,464	2,598
¾	1,576	6,304	4,728
1	2,459	9,836	7,377
1¼	4,263	17,052	12,689
1½	5,808	23,232	17,424
2	9,564	38,256	28,692
2½	13,623	54,492	40,869
3	21,041	84,104	63,123
3½	28,213	112,852	84,039
4	36,285	145,140	108,855
5	56,971	227,884	170,913
6	82,336	329,344	247,008

Formula:

$$\frac{A \times V \times 3600 \times 50}{144 \times 3.84} = \text{pounds per hour passed by orifice.}$$

where

A = area of inlet pipe, square inches.
V = velocity of steam through orifice (approximately 870 fps).
50 = 70 per cent efficiency of orifice less 20 per cent for factor of safety.
144 = square inches in 1 sq ft.
3600 = seconds in one hour.
3.8 = cubic feet per pound at 100-lb gage.

that the steam is flowing into a pressure less than 58 per cent of the initial pressure, that the orifice efficiency is approximately 70 per cent, and that 20 per cent more is allowed for a factor of safety, then the pressure reducing valves will have the working capacities shown in Table 10. If the valve, when fully open, does not give an orifice area equal to that of the pipe on the inlet side, then the capacities will be proportional to the percentage of opening secured, taking the pipe area as 100 per cent.

Most exact regulation of pressure on steam heating systems is secured from diaphragm-operated valves controlled by a pilot line from the low pressure pipe, taken off the low pressure main at least 15 ft from the reducing valve. The reducing valves operating on the proportional-reduction principle will give a variation of steam pressure on the low pressure side if the initial pressure varies between considerable limits. The so-called dead-end valve is used for reduced pressures where the line has not sufficient condensing capacity at all times to condense the leakage that might occur with the ordinary valve. Single-disc valves do not give as close regulation as double-disc valves, but the single disc is preferable where dead-end valves are necessary, such as on short runs to thermostatically controlled hot water heaters, central fan heating units and unit heaters.

The correct installation (Fig. 2) of a pressure-reducing valve includes a pressure-reducing valve with a gate valve on each side, a by-pass controlled by a globe valve, a pressure gage on the low pressure side, and a safety valve on the low pressure main at some point, usually within a reasonable distance of the pressure-reducing valve. Pressure-reducing valves should have expanded outlets for sizes greater than 2 in. Where the steam main is of still larger diameter than the expanded outlet, and in

cases where straight valves are used, an increaser is placed close against the outlet of the valve to reduce the velocity immediately after passing through the valve. Strainers are recommended on the inlets of all pressure-reducing valves. A pressure gage may be located on the high-pressure line near the valve if desired.

Owing to the large variation in steam demand on the average heating system, it is generally advisable to use two pressure-reducing valves connected in parallel. One valve should be large enough for the maximum load and the other should have a diameter approximately half that of the first. The smaller valve can be used most of the time, for it will give much better regulation than the larger one on light or normal loads.

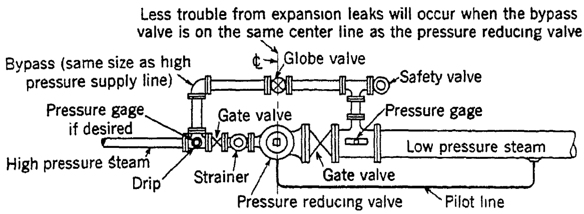


FIG. 2. TYPICAL PRESSURE-REDUCING VALVE INSTALLATION

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

EXPANSION IN STEAM AND RETURN LINES

Because all steam and return lines expand and contract with changes in temperature, provision should be made for such movement. The expansion in steam supply pipes is normally taken at 1¼ to 1½ in. per 100 ft and in return lines at one-half or two-thirds of this amount. It may be calculated accurately if the temperature rise and fall can be determined with reasonable certainty (Page 586, Chapter 34). The temperature at the time of erection often has a greater expansion effect on piping than the temperature in the building after it has been put into service.

Expansion may be taken care of by any, or all, of three different methods, namely, (1) the spring in the pipe including offsets and expansion bends, (2) the turning of the pipe on its threads and swing joints, and (3) the use of expansion joints.

By the first scheme, which is the most popular method where space permits, the pipe is offset, or *broken*, around rooms or corners, and is hung so that the spring in the pipe at right angles to the expansion movement is sufficient to absorb the expansion. If conditions do not lend themselves to this treatment, regular expansion bends of the *U* or offset type may be used. In tight places such as pipe tunnels the expansion joint is preferable.

On riser runouts and radiator runouts the swing joint is used almost without exception. On high vertical risers the pipes may be reversed every five to ten stories; that is, the supply is carried over to the adjacent return riser location and the return riser is run over to the former supply riser location, thus making horizontal offsets in each line. Corrugated copper expansion joints also are used on risers but must be made accessible in case future replacement becomes necessary.

EXPANSION BENDS

The calculation of the distance required for offsets and the size of expansion bends necessary to absorb a given amount of expansion leads into complicated formulae and is a subject of controversy. It seems to have been demonstrated, however, that the shape of the bend, the radius used, the relative amounts of straight and curved pipe in a bend, and the

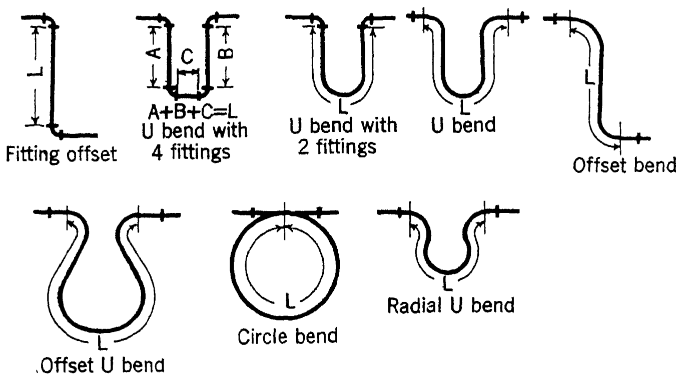


Fig. 3. Measurement of L on Various Pipe Bends and Offsets for Absorbing Expansion

type of bend have little bearing on the amount of expansion for which they will safely provide. The size, weight and material of the pipe and the length of all of the pipe in the bend, or even in the offset, have a bearing on its capacity to absorb expansion without straining the pipe material beyond the safe working stress. In Fig. 3 typical pipe bends and offsets for absorbing expansion are shown. The lengths L are those which are used in determining the stress in the pipe.

Fig. 4 shows a set of curves for standard weight steel pipe bends from which the approximate amount of pipe L (Fig. 3) for each pipe size may be determined from the amount of expansion movement that must be absorbed. These curves are such that the maximum fiber stress in any part of the bend will not be over 16,000 lb per square inch. Since 12,000 lb per square inch is considered to be a maximum working fiber stress in wrought iron pipe, an additional 33⅓ per cent must be added to the length of this type of pipe.

The amount of expansion can be doubled for a bend if the bend is cold sprung for one-half of the expansion movement. In other words, if the bend is erected with the main pipe cut short one-half of the expected expansion and the bend is then sprung open to meet the shortened pipe,

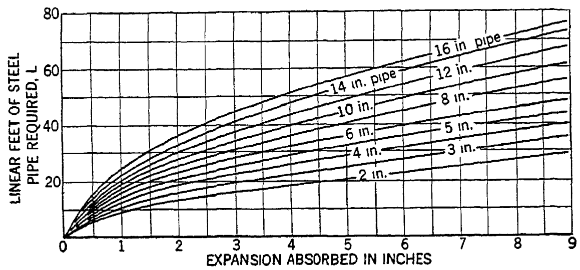


FIG. 4. CURVES GIVING LENGTH L OF BEND OF OFFSET NECESSARY TO ABSORB EXPANSION (WITHOUT COLD SPRING)

the expansion in the main will first allow the bend to go back to its neutral point and then will compress the bend an equal distance beyond the neutral point, thus securing a doubled capacity. Generally only a portion of the cold spring is considered as being effective, owing to the difficulties of erecting the bends with sufficient exactitude in the length of the main line and the difficulty of cold springing. See additional material on pipe expansion in Chapter 34.

PIPING CONNECTIONS AND DETAILS

Piping connections may be classified into two groups: *first,* those suitable for any system of steam heating; *second,* those devised for certain systems which cannot be satisfactorily applied to any other type. There are also various details that apply to piping on the steam side which cannot be used on the returns. An installation that is designed and sized correctly and installed with care may be rendered defective by the use of improper connections, such as runouts that do not allow for expansion, thermostatic traps unprotected from scale, pressure-reducing valves without strainers, and lack of drips at required points.

BOILER CONNECTIONS

Supply

Boiler headers and connections have the largest sizes of pipe used in a system. Cast-iron, horizontal-type, low pressure heating boilers usually have several tapped outlets in the top, the manufacturers recommending their use in order to reduce the velocity of the steam in the vertical uptakes from the boiler and to permit entrained water to return to the boiler instead of being carried over into the steam main where it must be cared for by dripping. Steel heating boilers usually are equipped with only one steam outlet but many engineers believe that better results are obtained by specifying that such boilers have two. The second outlet, usually located 3 or 4 ft back of the regular one, reduces the velocity 50 per cent in the steam uptake.

Fig. 5 shows a type of boiler connection that was used for many years and one with which some boilers are now piped. The uptakes are carried as high as possible, turned horizontally and run out to the side of the boiler and then are connected together into the main boiler runout which

drops into the top of the boiler header through a boiler stop valve. No drips are provided on this type of runout except a very small one which is sometimes installed on the boiler side of the stop valve. Fig. 6 shows a type of boiler connection which is regarded as superior to that shown in Fig. 5 and which is the type illustrated in the system diagrams in Chapter 31. This type is similar to that shown in Fig. 5 except that the horizontal branches from the uptakes are connected into the main boiler runout, and the steam is carried toward the rear of the boiler. The branch to the building or boiler header is taken off *behind* the last horizontal boiler connection. At the rear end of this main runout, a large size drip, or balance pipe, is dropped down into the boiler return, or into the top of the Hartford Loop, which is described in a following paragraph. As a result, any water carried over from the boiler follows the direction of steam flow

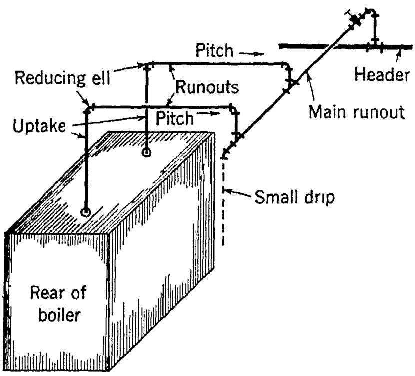


FIG. 5. OLD STYLE STANDARD BOILER CONNECTIONS

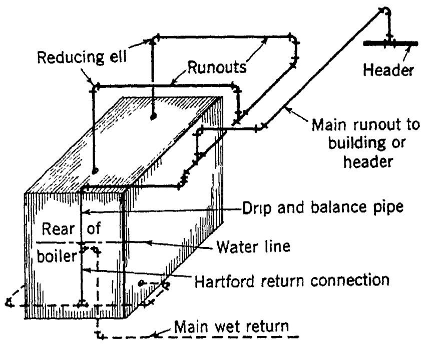


FIG. 6. APPROVED METHOD OF BOILER CONNECTIONS

toward the rear and is discharged into the rear drip, or balance pipe, without being carried over into the system.

Return

Cast-iron boilers are generally provided with return tappings on both sides, but steel boilers often are equipped with only one return tapping. A boiler with side return tappings will usually have a more effective circulation if both tappings are used. Check valves generally should not be used on the return connection to steam heating boilers because they are not always dependable inasmuch as a small piece of scale or dirt lodged on the seat will hold the tongue open and make the check useless. These valves also offer a certain amount of resistance to the returns coming back to the boiler, and in gravity systems will raise the water line in the far end of the wet return several inches[3]. However, if check valves are omitted and the steam pressure is raised with the boiler steam valve closed, the water in the boiler will be blown out into the return system with the accompanying danger of boiler damage. These objections are largely overcome with the Hartford return connection.

[3]See method of calculating height above water line for gravity one-pipe systems in Chapter 31.

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford Connection, or the Underwriters Loop, is recommended. Fig. 7 shows this connection for both single boiler and two-boiler installations. By balancing the column of water in the loop against the steam pressure, the water cannot be blown out of the loop whatever the relative pressure conditions in the boiler, steam lines, or return lines. This balancing is done by raising the return to approximately the normal water line of the boiler, looping it back to the boiler inlet and connecting the top of this loop by means of a balance pipe with the steam runout from the boiler. It is important that this balance pipe be connected into the boiler steam line on the boiler side of all valves.

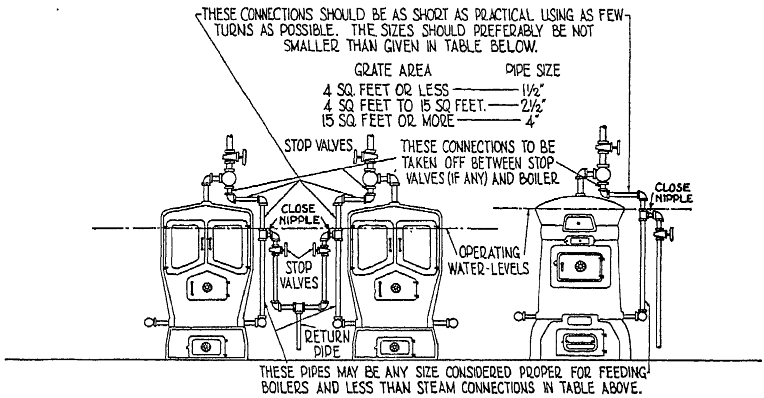


FIG. 7. THE HARTFORD RETURN CONNECTION

Theoretically, the top of the loop should be at the normal boiler water line but since this installation often causes trouble from water hammer in the top of the loop, this top is usually made 2 in. below the normal boiler water line to keep the horizontal pipe at the top submerged under all normal conditions. It is important that this top of the loop be made with the shortest possible horizontal pipe, a close nipple being employed.

Sizing Boiler Connections

Little authentic information is available on the sizing of boiler runouts and steam headers. Although many engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation, and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the

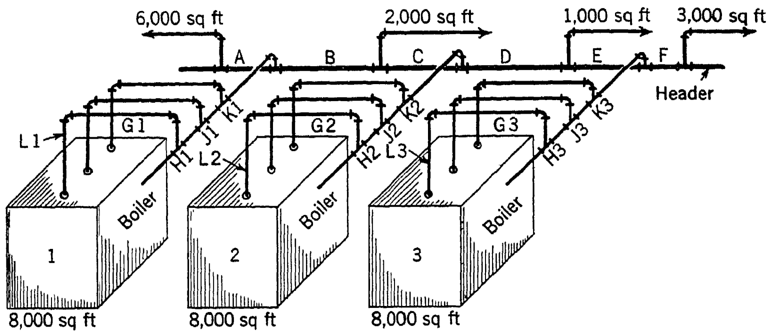


Fig. 8. Boiler Steam Header and Connections

boiler and the horizontal main or runout is compensated for by the use of reducing ells (Figs. 5 and 6).

The following example illustrates the sizing of the boiler connections shown in Fig. 8.

Example 4. Determine the size of boiler steam header and connections (Fig. 8) if there are three boilers, two to carry 50 per cent of the load each, and the third to be used as a spare. The steam mains are based on ⅛-lb drop per 100 sq ft of equivalent direct radiation (EDR).

Solution:

Size of Boiler Header

When Operating on Boilers	Load on Various Portions of Header						Maximum Load
	A	*B*	*C*	*D*	*E*	*F*	
Nos. 1 and 2	6000	0	2000	4000	3000	3000	6000
Nos. 2 and 3	6000	6000	8000	2000	3000	3000	8000
Nos. 3 and 1	6000	0	2000	2000	3000	3000	6000
Max. Load	6000	6000	8000	4000	3000	3000	8000

8000 sq ft @ ⅛ lb per 100 ft = 6 in. main. (See Table 7.)

Size of Boiler Runouts

The three runouts

$G_1, G_2, G_3 = \frac{8000}{3} = 2667$ sq ft each @ ⅛ lb per 100 ft = 4 in. pipe.

$H_1, H_2, H_3 = 2667$ sq ft each @ ⅛ lb per 100 ft = 4 in. pipe[4] (See Table 7).

$J_1, J_2, J_3 = 5333$ sq ft each @ ⅛ lb per 100 ft = 5 in. pipe[4] (See Table 7).

$K_1, K_2, K_3 = 8000$ sq ft each @ ⅛ lb per 100 ft = 6 in. pipe[4] (See Table 7).

The uptakes from the boiler probably would be 6 in. pipe with a 6 in. × 4 in. reducing ell at top.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to

[4] *Note.*—As K_1, K_2, K_3 all carry 8000 sq ft and are 6 in. pipe, the whole runout including J_1, J_2 and J_3 would be made 6 in. pipe, also.

two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Friction Losses for Various Rates of Flow of Water, Fig. 3, Chapter 35. The relative boiler loads should be considered, as in the case of gravity return connections.

Radiator Connections

Radiator connections are important on account of the number of repetitions which occur in every heating installation. They must be properly pitched and they must be arranged to allow not only for movement in the riser but, in frame buildings, for the shrinkage of the building. In a three story building this sometimes amounts to 1 in. or more. The simplest connection is that for the one-pipe system where only one radiator connection is necessary. Where the radiator runouts are located on the ceiling or under the floor, sufficient space usually is available to make a good swing joint with plenty of pitch, but where the runouts must come above the floor the vertical space is small and the runouts can project out into the room only a short distance. Fig. 9 illustrates two satisfactory methods of making runouts on a one-pipe gravity air vent system of either the up-feed or down-feed type, the runout below the floor being indicated in full lines and the runout above the floor in dotted lines. Sometimes it is necessary to set a radiator on pedestals, or to use high legs, in order to obtain sufficient vertical distance to accommodate above-the-floor runouts. Particular attention must be given to the riser expansion as it will raise the runout and thereby reduce the pitch.

Similar connections for a two-pipe system of the gravity air vent type are illustrated in Fig. 10 for the old steam type radiator. If the water type is used, the supply tapping is at the top instead of at the bottom, the runouts otherwise remaining as shown in Fig. 10. A satisfactory type of radiator connection for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of both the up-feed and down-feed types is shown in Fig. 11.

While short radiators, not exceeding 8 to 10 sections, may be supplied and returned from the same end as indicated in Fig. 12, the top-an-bottom-opposite-end method is to be preferred in all cases where it can be used. On down-feed systems of the atmospheric, vapor, vacuum, sub-atmospheric, and orifice types, the bottom of the supply riser must be dripped into the return somewhat as illustrated in Fig. 13. On up-feed systems of the vapor and atmospheric types, where radiators in the

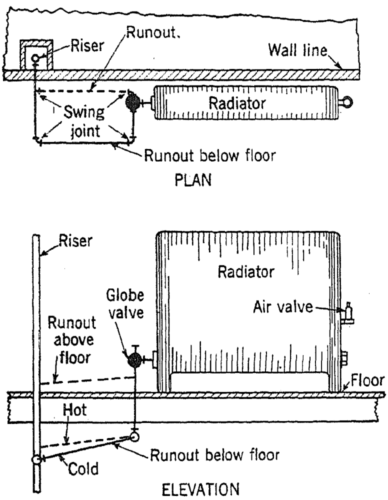


FIG. 9. TYPICAL ONE-PIPE RADIATOR CONNECTIONS (UP-FEED OR DOWN-FEED)

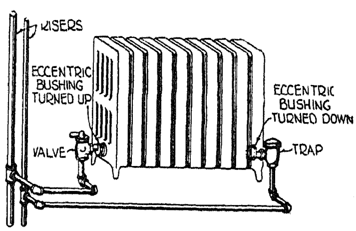


FIG. 10. CONNECTIONS TO STEAM-TYPE RADIATOR FOR TWO-PIPE GRAVITY SYSTEM, UP-FEED OR DOWN-FEED

Note.—Steam-type radiators should not be used on any except gravity one-pipe and gravity two-pipe systems.

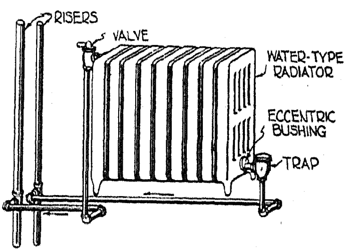


FIG. 11. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS FROM UP OR DOWN-FEED RISERS

Note.—Suitable for up-feed or down-feed atmospheric, vapor, vacuum, sub-atmospheric and orifice systems.

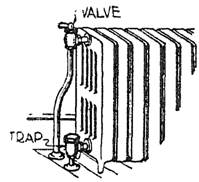


FIG. 12. TOP AND BOTTOM RADIATOR CONNECTIONS FROM UP- OR DOWN-FEED RISERS. (NOT TO EXCEED 8 TO 10 SECTIONS.)

Note.—Suitable for up-feed or down-feed atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems. Opposite end connections always preferable.

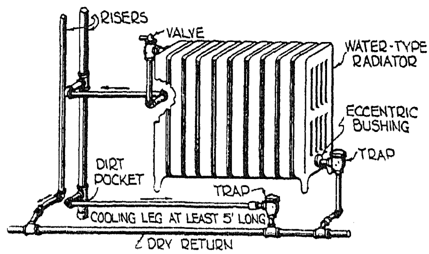


FIG. 13. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS WITH HEEL OF DOWN-FEED RISER DRIPPED INTO DRY RETURN

Note.—Suitable for down-feed only. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

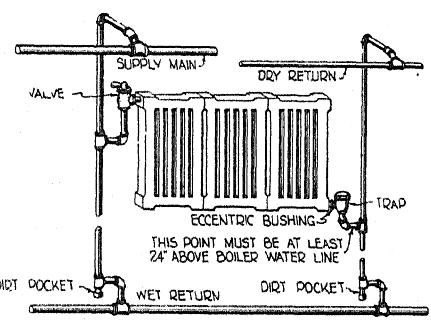


FIG. 14. CONNECTIONS TO RADIATOR HUNG ON WALL

Note.—For up-feed with radiators below level of steam main. For atmospheric and vapor systems. Not suitable for vacuum, sub-atmospheric, or orifice systems.

basement are located below the level of the steam main, the drop to the radiator is dripped into the wet return and an air line is used to vent the return radiator connection into an overhead return line, as illustrated in Fig. 14. When the radiator stands on the floor below the main, the drip on the steam branch down to the radiator may be omitted if an overhead valve, as shown in Fig. 15, is used. This method is also suitable for vacuum, sub-atmospheric, and orifice systems.

Convector Connections

Convectors often are installed without control valves, a damper being used to shut off the flow of air to retard the heat transfer from the convector even though it is still supplied with steam. The piping connections for a convector with the inlet and outlet at the same end are shown in Fig. 16. There is no valve on the steam side but there is a thermostatic trap on the return. The damper for control is shown immediately above the convector. This piping is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of the up-feed type. A similar unit with connections on opposite ends and suitable for the same systems is shown in Fig. 17. This unit has no damper but requires a valve on the steam connection for control. When valves must be located so as to be accessible from the supply air grille, the arrangement usually takes the form indicated in Fig. 18. Convectors with damper control, installed in cabinets or under window sills, usually are connected as shown in Fig. 19. A convector located in the basement and supplying air to a room on the floor above may be piped as pictured in Fig. 20 for all systems except gravity one-pipe or two-pipe systems.

Vapor systems with heating units in the basement where the returns are wet would be treated as in Fig. 21. Similar heating units where a dry return is available would be connected as shown in Fig. 22. If the dry return were on a vacuum, atmospheric, sub-atmospheric or orifice system, the treatment would be identical.

Pipe Coil Connections

Pipe coils, unless coupled in a correct manner, often give trouble from short circuiting and poor circulation. The method of connecting shown in Fig. 23 is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Indirect Air Heater Connections

Heating units for central fan systems have simple connections on the steam side. The steam main is carried into the fan room and has a single branch tapped off for each row of heating units. Each of these main branches is split into as many connections as need be made to each row, governed by the number of stacks and the width of the stacks. Each stack must have at least one steam connection, and wide stacks are more evenly heated with two steam connections, one at each end.

The piping shown in Fig. 24 is for small stacks and has the steam connected at only one end. On the return side all of the returns are collected together through check valves and are passed through blast traps which are connected to the vacuum return or to an atmospheric return. The air

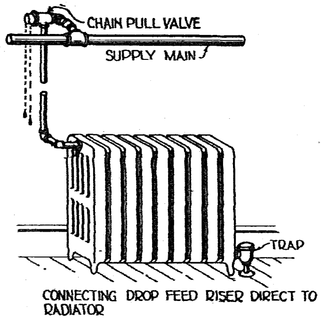


FIG. 15. CONNECTING DROP FEED RISER DIRECT TO RADIATOR BY TURNING VALVE ON ITS SIDE

Note.—Suitable for up-feed with radiators below level of steam main. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

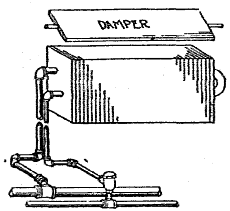


FIG. 16. CONVECTOR CONNECTIONS SAME END

Note.—Suitable for up-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

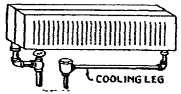


FIG. 17. HORIZONTAL SECTIONAL FIN-TYPE HEATING UNIT

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

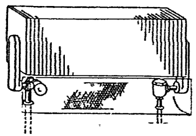


FIG. 18. FIN-TYPE HEATING UNIT VALVES BEHIND GRILLE

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

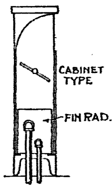


FIG. 19. FIN-TYPE HEATING UNIT CONCEALED IN CABINET

Note.—Suitable for any two-pipe system.

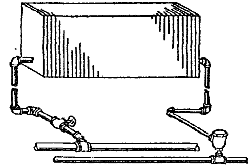


FIG. 20. FIN-TYPE HEATING UNIT WITH VALVES IN BASEMENT

Note.—Suitable for up-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

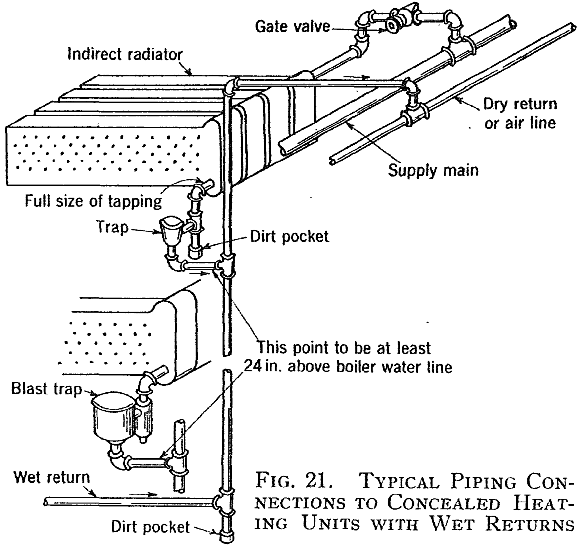


FIG. 21. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

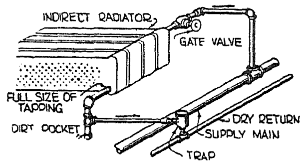


FIG. 22. TYPICAL PIPING CONNECTIONS TO INDIRECT RADIATORS WITH DRY RETURN

Note.—Suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

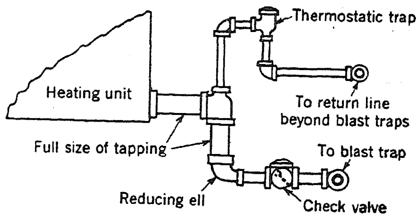


FIG. 25. HEATING UNIT RETURN CONNECTION WITH SEPARATE AIR LINE

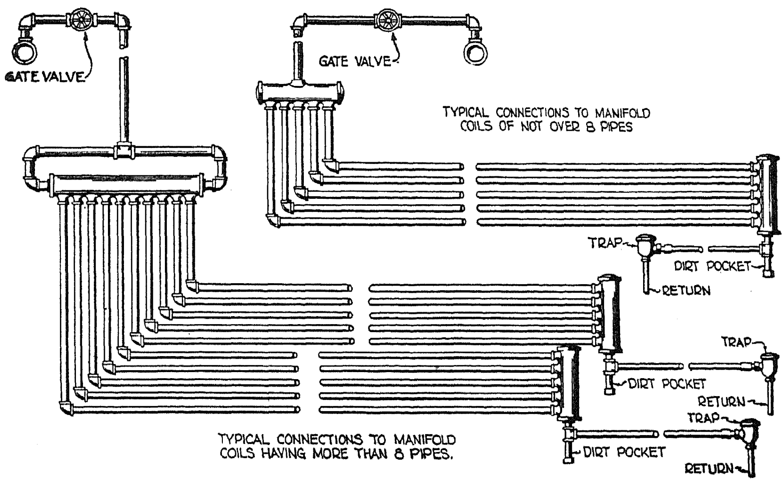


FIG. 23. TYPICAL PIPE COIL CONNECTIONS

Note.—Suitable for up-feed or down-feed. For **atmospheric, vapor, vacuum, sub-atmospheric, and** orifice systems.

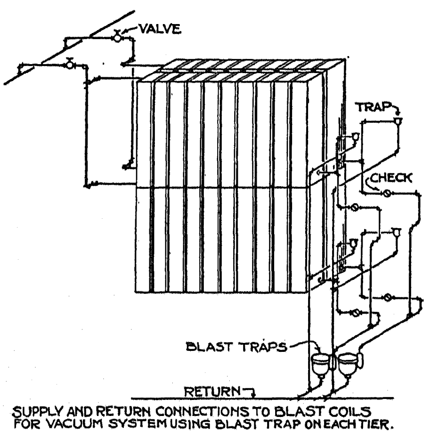


FIG. 24. CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

Note.—Suitable for atmospheric and vacuum systems.

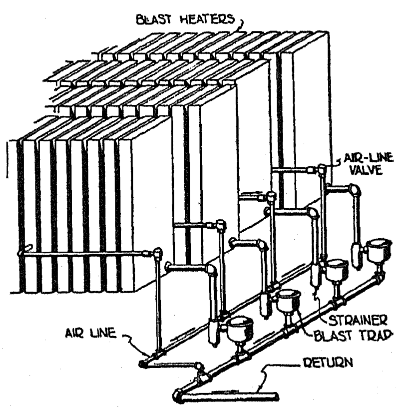


FIG. 26. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

Note.—Suitable for vacuum and atmospheric systems.

from the stacks, in the case illustrated, passes up into a small air line and through a thermostatic trap into a line connecting into the return beyond the blast trap. It is important to use a nipple the full size of the outlet tapping on the stack and to reduce the pipe size to the normal return size required, by the use of a reducing ell, as indicated in Fig. 25.

Where the stacks contain some thirteen or more sections, an auxiliary air tapping is made to the lower portion of one of the middle sections, in the manner illustrated in Fig. 26, to prevent air collecting at this point. Thermostatic control as applied to such heating units in modern practice

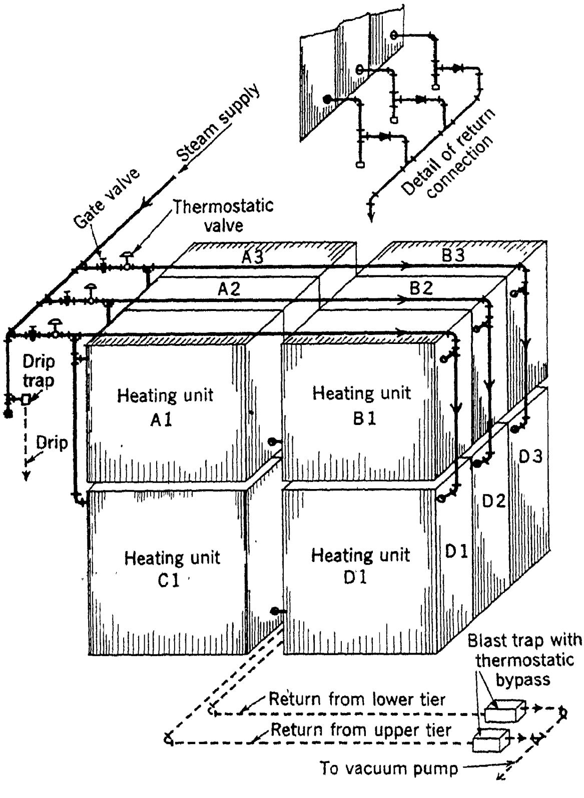


FIG. 27. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

consists of a thermostatic valve located in each main branch from the steam line so that each valve will open or close a complete row of stacks across the entire face of the heating unit. The stack closest to the fresh air intake is not usually equipped with a control valve. Steam is furnished continually to this coil to prevent freezing, and only the supply pipe is equipped with a gate valve. In this case no particular attention need be paid to the method of connecting the returns, that is, they do not need to be connected in parallel with the steam connections but may be hooked together in any convenient manner. The arrangement shown in

Fig. 27 is satisfactory. A detail of the arrangement where a connection is made with a stack is shown in Fig. 28. It is essential to have a check valve on each individual stack to prevent reverse flow when the thermostatic valve in the steam line closes off and a partial vacuum is produced in the stack. The end of the steam main also should be dripped as indicated in Fig. 27.

If the separate air line is used as shown in Fig. 24, the blast traps may be supplied without thermostatic by-passes but if the piping is arranged as shown in Figs. 26 or 27, the blast traps must be supplied with the thermostatic by-passes to permit the passage of the air.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_l - t_e)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \tag{3}$$

where

EDR = equivalent direct radiation, square feet.
Q = volume of air, cubic feet per minute.
t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.
t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.
60 = the number of minutes in one hour.
55.2 = the number of cubic feet of air heated 1 F by 1 Btu.
240 = the number of Btu in 1 sq ft of EDR.

Example 5. Assume that the heating units shown in Fig. 27 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.
[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

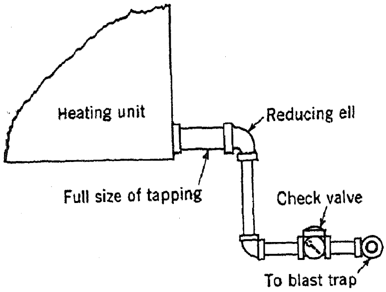


FIG. 28. HEATING UNIT RETURN CONNECTION WITHOUT SEPARATE AIR LINE (CENTRAL FAN SYSTEM)

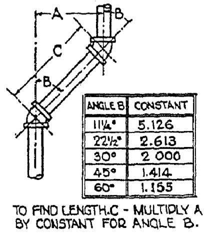


FIG. 32. CONSTANTS FOR DETERMINING PROPER LENGTH OF OFFSET PIPE

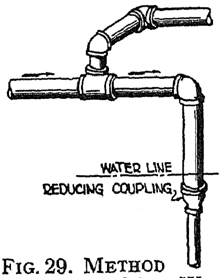


FIG. 29. METHOD OF DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

Note.—Suitable for vapor and atmospheric systems.

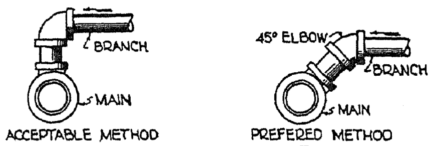


FIG. 33. ACCEPTABLE AND PREFERRED METHODS OF TAKING BRANCH FROM MAIN

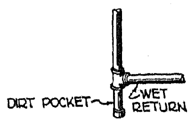


FIG. 34. DIRT POCKET CONNECTION

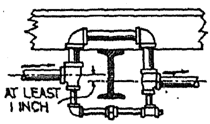


FIG. 30. LOOPING MAIN AROUND BEAM

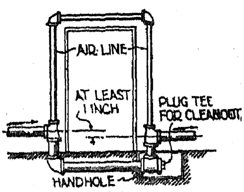


FIG. 31. LOOPING DRY RETURN MAIN AROUND OPENING

Note.—Suitable for any dry return line and any return line carrying air.

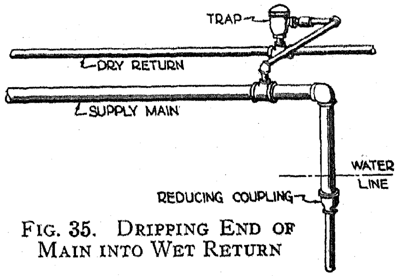


FIG. 35. DRIPPING END OF MAIN INTO WET RETURN

Note.—Suitable for vapor systems.

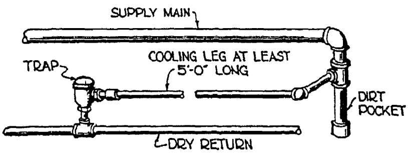


FIG. 36. DRIPPING END OF MAIN INTO DRY RETURN. (A GATE VALVE IS RECOMMENDED AT THE INLET SIDE OF THE TRAP)

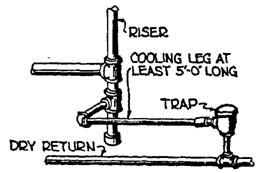


FIG. 37. DRIPPING HEEL OF RISER INTO DRY RETURN. (A GATE VALVE IS RECOMMENDED AT THE INLET SIDE OF THE TRAP)

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

DRIPPING

Any steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the steam flow. Any steam main in any heating system can be elevated if dripped (Fig. 29). Steam mains also may be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 30). Return mains may be carried past doorways or other obstructions by using the scheme illustrated in Fig. 31; in vacuum systems it is well to have a gate valve in the air line.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 32.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 33, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 34.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 35. On vacuum systems the ends of the steam mains are dripped and vented into the return through thermostatic drip traps opening into the return line. The same method may be used in atmospheric systems. The cooling leg (Fig. 36) is for cooling the condensation

sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 37.

PROBLEMS IN PRACTICE

1 ● What is the equivalent length of run in a steam system?

It is the length of straight pipe which will have the same friction and pressure drop as a shorter length of pipe of the same size with accompanying valves, tees, elbows, and other fittings will have, when both pipes are carrying the same amount of steam at the same pressure.

2 ● When the size of pipe is still undetermined, what arbitrary percentage is usually added to the actual length to obtain the equivalent length?

Usually 100 per cent; in other words, the actual length is doubled to allow for the added drop produced by the valves, tees, elbows, and other fittings.

3 ● What are the major factors to be considered in determining the flow of steam in pipes?

a. The initial steam pressure available and the total pressure drop allowable between the source of steam supply and the end of the return system. The pressure drop should never exceed one half of the initial pressure.

b. The maximum steam velocity allowable. When condensate is flowing against the steam, the velocity must not be so great as to produce water hammer, or hold up water in parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which disturbances take place depends upon:

1. Size of pipe.
2. Whether pipe is vertical or horizontal.
3. Pitch or grade of pipe.
4. Quantity of water flowing against steam.

c. The equivalent length of run from the source of steam supply to the farthest heating unit, with allowance for friction in pipe fittings and valves.

4 ● Name three fundamental considerations in designing the piping system for steam heating.

a. Provision for the distribution of suitable quantities of steam to the various heating units.

b. Provision for the return of condensate from the radiators and piping to the boiler.

c. Provision of means for expelling air from the radiators and piping.

5 ● Why is the proper reaming of the ends of pipe necessary?

The capacities of pipes depend upon the free area available for flow. In cutting the pipe this area may be restricted by a burr, which may decrease the capacity of a pipe more than 25 per cent in the smaller pipe sizes.

6 ● a. What are the major factors to be considered when selecting a pressure reducing valve?

b. How should such valve be installed?

a. The initial pressure of the steam must be considered along with the desired reduced pressure. The connected load to be supplied must be known in square feet of equiva-

lent direct radiation or in pounds of steam per hour. For operation with a continuous load, a semi-balanced or double seated valve operated by a diaphragm gives good results. Where the load is intermittent, as in process work or with thermostatically controlled blast heaters, a so-called dead end or single seated valve should be used.

The pressure reducing valve should be installed in a horizontal line with a gate valve on each side, and with a by-pass operated by a valve. The pressure balancing pipe from the diaphragm chamber should be connected into the top or side of the low pressure main not less than 15 ft from the reducing valve.

7 ● What is the usual expansion allowance and how it is compensated for in heating system supply risers?

The expansion of low pressure steam piping is normally taken as 1¼ to 1½ in. per 100 ft of pipe. With a five story building a double swing connection between the riser and the main will suffice. In buildings between 5 and 10 stories high the riser should be anchored near its center and have double swing connections to the main. For taller buildings expansion loops or riser offsets are used which are capable of handling a length of riser reaching 5 stories in either direction from the joint. The risers are anchored at each alternate 5 stories. All radiators must have double swing connections, and those connected above where the riser is anchored must be given greater pitch to insure their having proper grade when the riser is heated.

8 ● Why should all boiler steam supply tappings be used full size?

In order to operate at low steam velocities so the water in suspension can separate from the steam and remain in the boiler.

9 ● What is the Underwriters Loop or the Hartford Connection?

An arrangement of piping on the returns to low pressure boilers wherein the return line is raised up nearly to the water line of the boiler and is then dropped back and connected to the boiler return inlet; the high point is connected by a balance pipe to the steam runout from the boiler on the boiler side of all stop valves. With this loop no check valve is required, and water cannot be backed out of the boiler and into the return at a point lower than the invert of the pipe at the top of the loop.

10 ● What are the important factors in making radiator connections?

Connections to radiators should be made as direct as possible, of proper size, with ample pitch of piping and allowance for expansion.

11 ● Why should careful attention be given to proper dripping and drainage of steam piping?

The steam mains and risers must be quickly drained of condensate and where necessary vented of air in order to obtain a sufficient supply of steam to the radiators. Proper drainage is also necessary to insure a noiseless heating system.

12 ● What is the limit of pressure drop usually recommended in a vacuum system?

Not over ⅛ lb (2 oz) per 100 ft of equivalent run, and not over 1 lb total drop.

13 ● When steam and condensation are flowing in the same direction, what is the maximum total pressure drop which should be used?

The maximum total pressure drop should not exceed one half of the initial steam pressure.

14 ● What does a proper installation of a pressure reducing valve include?

A strainer in front of the pressure reducing valve; a gate valve in front of the strainer; a gate valve after the reducing valve; a by-pass around the two gate valves, strainer, and pressure reducing valve; and a globe valve in the by-pass. Sometimes a safety valve on

the low pressure side and pressure gages on both sides are installed. The high pressure line should be dripped just before the high pressure steam enters the pressure reducing valve assembly.

15 ● Will a pressure reducing valve which is reducing the steam pressure from 100 lb gage to 50 lb gage pass more or less steam than the same valve when reducing the steam pressure from 100 lb gage to 5 lb gage?

The valve will pass practically the same volume of steam in each case as the velocity of steam flowing through an orifice shows no material increase after the reduced absolute pressure has fallen to 58 per cent of the initial absolute pressure. Because of its greater density, the weight of steam passed will be greater in the case of the reduction to 50 lb gage.

Chapter 33

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Selecting Pipe Sizes, Forced Circulation, Effect of Variations in Pipe Sizes, Gravity Circulation, Mechanical Circulation, Expansion Tanks, Installation Details

A HOT water heating system is one in which water is the medium by which heat is carried through pipes from the boiler to the heating units. There are two general types, namely, *forced circulation* and *gravity circulation* systems. In the former the pressure head maintaining flow is produced mechanically, whereas in the latter the pressure head is produced by the differences in weight of the water in the flow and in the return risers.

The fundamental rule in the design of a hot water system is that the total friction and resistance head in any circuit must equal the pressure head causing the water to flow in the same circuit.

In designing a hot water heating system, it is necessary to determine:

1. The heat losses of the rooms or spaces to be heated. (See Chapter 7.)
2. The size and type of boiler. (See Chapter 25.)
3. The location, type, and size of heating units. (See Chapter 30.)
4. The method of piping.
5. Suitable pipe sizes.
6. The type and size of circulating pump (if forced circulation).
7. The type and size of expansion tank.

The unit, a square foot of equivalent direct radiation, EDR, has been used for many years for rating purposes in both steam and hot water systems, but its use, especially in hot water systems, has always resulted in complications and confusion. It is the plan of The Guide *to eventually eliminate this empirical expression and to substitute a logical unit based on the Btu. The* Mb, *the equivalent of 1000 Btu, and the* Mbh, *the equivalent of 1000 Btu per hour, which have been approved by the A.S.H.V.E., are used in this chapter on hot water systems to replace the square foot of radiation formerly used.*

ONE- AND TWO-PIPE SYSTEMS

Pipe systems may be divided into two general types, namely, *two-pipe* and *one-pipe* systems. In a two-pipe system the piping is arranged so that the water flows through only one radiator during a circuit through the system, so that all radiators are supplied with water at practically the same temperature as that in the boiler. In a one-pipe system, the water flows through more than one radiator during its circuit. In that case, the

first radiator receives the hottest water; the second radiator, somewhat cooler water; the third one, still cooler; and so on. As the temperature of the water supplied to a radiator is lowered, the size of the radiator must be increased and, consequently, the total heating surface for a one-pipe system must be greater than that for a two-pipe system for the same service.

Two-pipe systems may be divided into two classes, *direct return* systems (Fig. 1), and *reversed return* systems (Fig. 2). In a direct return system the water returns to the heater by a direct route after it has passed through its radiator and, as a result, the paths through the three radiators shown in Fig. 1 are of unequal lengths, the path through the first radiator being the shortest and that through the third radiator, the longest. In a reversed return system, the water returns to the heater by an indirect route after it has passed through the radiators, so that the paths leading through the three radiators shown in Fig. 2 are practically of equal length.

The reversed return system has an advantage over the direct return system in that it is more likely to function satisfactorily even though the

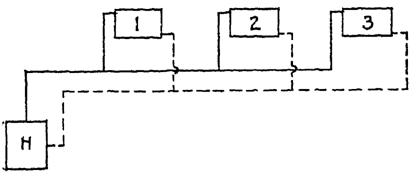


FIG. 1. A DIRECT RETURN SYSTEM

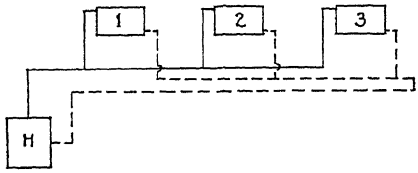


FIG. 2. A REVERSED RETURN SYSTEM

pipe system is not accurately designed. For example, if in Fig. 2 all pipes are of one size, each of the three radiators will receive approximately the same quantity of hot water because the three paths are practically of equal length, whereas in Fig. 1, if all pipes are of the same size, Radiator 1 will receive more water than the others because the path through it is shorter than those through the other radiators. As a result, Radiator 1 will be filled with water at a higher average temperature than the remaining two radiators, and will therefore dissipate more heat. To prevent this unequal distribution of heat it is necessary to throttle the paths through Radiators 1 and 2 so that the friction heads of the three paths are equal when each radiator receives its proper quantity of water.

A comparison of Fig. 1 and Fig. 2 may suggest that a reversed return system requires considerably longer mains than a direct return system. This is not always the case. For example, note the reversed return system of Fig. 3.

PIPE SIZES

The pressure heads available in forced circulation systems are much larger than those in gravity circulation systems, consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipes of a heating system are reduced in size, the necessary increase in

the velocity of the water increases the cost of operating the circulating pump. There is an optimum velocity of the water in a heating system for which the sum of the cost of the system and the cost of its operation is a minimum. This velocity should be determined by calculation for the particular system under consideration.

Since the velocities in forced circulation systems are higher than those in gravity circulation systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system and, consequently, it

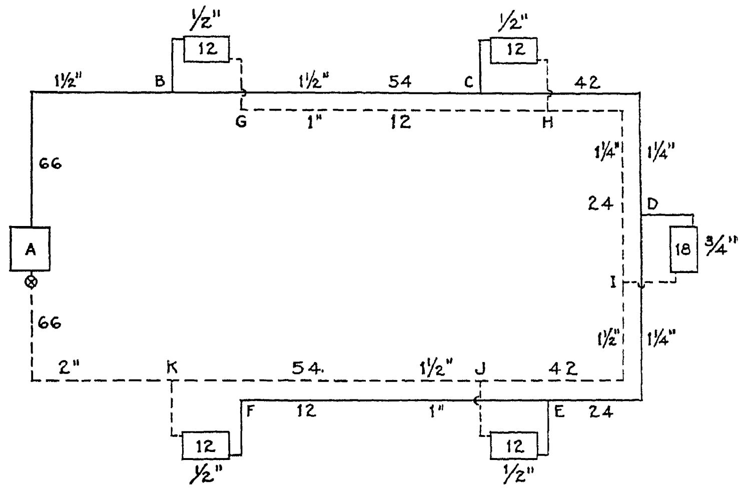


FIG. 3. A FORCED CIRCULATION REVERSED RETURN SYSTEM[a]

[a]This system could be divided into two branches. This would permit the use of smaller pipes and would produce only slight changes in the total length of the pipe. It is shown as a single system here simply to illustrate the method of determining pipe sizes by means of pipe size tables. Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet.

is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FORCED CIRCULATION

The following examples will illustrate the procedure to be followed in designing forced circulation systems:

Example 1. Assume that the paths through the five radiators shown in Fig. 3 consist each of 150 ft of mains, 5 ft of radiator connections, 1 boiler, 1 radiator, 1 radiator valve, 10 ells, and 2 tees. Design the piping for this system.

Solution. The friction heads in the boiler, radiator, valve, and tee may be expressed in terms of the friction head in one elbow according to the values given in Table 1. Having done this, each of the five circuits is taken as 155 ft of pipe and about 24 elbow equivalents. The friction head of one elbow is approximately equivalent to that in a pipe having a length equal to 25 diameters. Assuming that the average pipe size in this case will be about 1¼ in., one elbow equivalent may be placed equal to about 3 ft of pipe and the total length of the circuit equivalent to about 227 ft of pipe.

Having determined the equivalent pipe length, assume the rate at which the water is

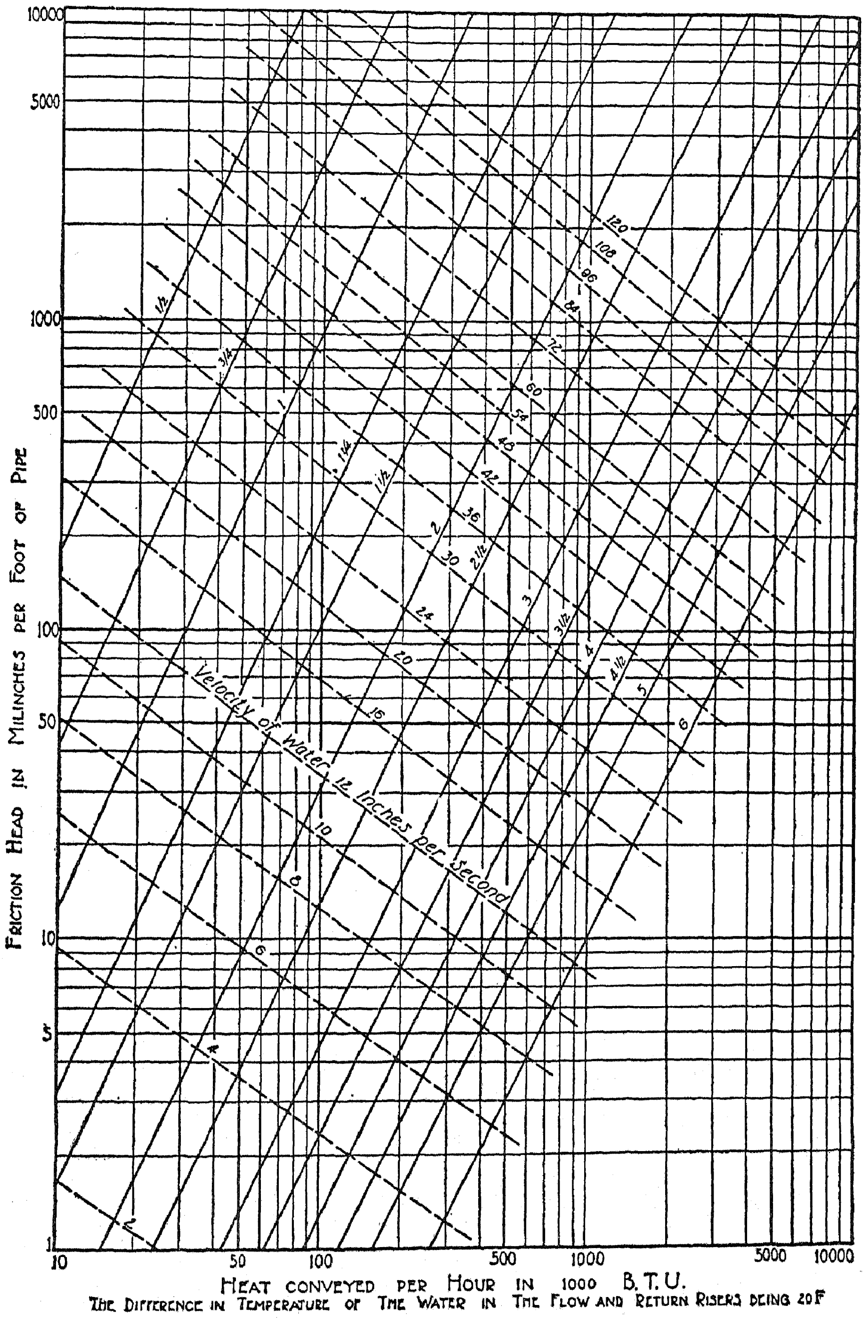


Fig. 4. Friction Heads in Pipes for a 20 F Temperature Difference of the Water in the Flow and Return Lines

to be forced through the system. This rate may vary widely. The water may flow through the radiator so that it will cool 10 deg or 20 deg or any other reasonable number of degrees. In this case, assume a 10-deg drop. Since the system is to dissipate 66,000 Btu per hour (66 Mbh), the pump must circulate 6600 lb of water per hour or 13.8 gpm based on the actual density of water of 7.99 lb per gallon at 215 F. One gallon of water per minute at this density will deliver 9600 Btu per hour (9.6 Mbh) with a temperature drop of 20 F.

TABLE 1. ELBOW EQUIVALENTS[a]

1 90-deg elbow	1.0
1 45-deg elbow	0.7
1 90-deg long turn elbow	0.5
1 open return bend	1.0
1 open gate valve	0.5
1 open globe valve	12.0
1 angle radiator valve	2.0
1 radiator	3.0
1 heater	3.0
1 tee	(Note[b])

[a]The loss of head in one elbow can be expressed in terms of the velocity head by the formula:

$$h = \frac{v^2}{2g} \tag{1}$$

where

h = the loss of head in feet, v = the velocity of approach in feet per second, and $2g$ = 64.4 ft per second per second.

[b]The loss of head in tees when water is diverted at right angles through a branch of the tee varies with the per cent diverted. When the water diverted is less than 60 per cent of that approaching the tee, the loss of head, in elbow equivalents, may be expressed as follows:

$$h_e = \frac{v_1^2}{v_2^2} \tag{2}$$

where

h_e = the loss of head in elbow equivalents, v_1 = the velocity of approach, v_2 = the velocity of water diverted at right angles.

Values in elbow equivalents for the most common percentages of water diverted in a 1x1x1-in. tee are as follows:

25%	16.0
33%	9.0
50%	4.0
100%	1.8

For other percentages the approximate values may be secured by interpolation. When the water is diverted from the tee into a smaller size branch, as in a 1x1x¾-in. tee, approximate values may be secured by means of Formula 2.

The next step in the design is to assume the velocity at which the water is to circulate through the system. This also may vary materially. As the velocity is increased, the sizes of the pipes and the cost of the system are decreased, but the cost of operating the circulating pump is increased. The designing engineer should make a careful study to determine the velocity which will produce the most economical installation for the particular case in hand. In this case, assume a velocity of about 1½ fps for a 1¼-in. pipe.

Reference to Fig. 4 shows that for a 1¼-in. pipe and a velocity of 18 in. per second, the friction head is about 100 milinches per foot, or about 2 ft for a circuit of 227 ft, if the pipe sizes for that circuit are chosen so that the average friction head is about 100 milinches per foot of pipe.

The pipe sizes may now be selected from Fig. 4 by making allowance for the fact that Fig. 4 is based on a temperature drop of 20 F and that the system to be designed is to have a temperature drop of only 10 F as follows: Sections *AB* and *KA* carry 66,000 Btu per hour (66 Mbh) with a temperature drop of 10 F; if the temperature drop were 20 F these sections would, with the same velocity and the same friction head, carry 132,000 Btu per hour (132 Mbh). Hence, refer to Fig. 4 for 132,000 Btu and a unit friction head of 100 milinches, and note that the correct size would be about halfway between a 1½-in. and a 2-in. pipe. Therefore, select a 1½-in. pipe for Section *AB* and

a 2-in. pipe for Section *KA*. The pipe sizes for the remaining eight sections and for the radiator connections can be selected in the same manner and recorded on the pipe diagram as shown.

The circulating pump for the system should be one which has its highest efficiency when it is delivering 13.8 gpm against a head of 2 ft.

If a number of heating systems are to be designed for similar conditions, *i.e.*, for a total friction head of 2 ft and a temperature drop through the radiators of 10 F when the maximum quantity of heat is being delivered to the building, a table such as Table 2 may be prepared from the data of Fig. 4. Having this table, the pipe sizes for the system of Example 1 can be easily selected. For example, for Sections *BC* and *JK*, each supplying 54 Mbh, the equivalent pipe length of the system is 227 ft. In the table the length shown nearest to this length is 200 ft. In the 200-ft column, a 1½-in. pipe is slightly too small and a 2-in. pipe is too large. The 1½-in. pipe will therefore be selected. For Sections *CD* and *IJ*, supplying 42 Mbh, a 1¼-in. pipe is too small and a 1½-in. pipe is too large, so 1¼ in. will be selected for the flow and 1½ in. for the return line. For larger

TABLE 2. CAPACITIES OF PIPES IN *Mbh* (1000 BTU PER HOUR) AND **Velocities of Water in Pipes in Inches per Second** FOR FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 2 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F[a]

1	2	3	4	5	6	7	8	9
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit						
		100	150	200	250	300	350	400
		Unit Friction Head, in Milinches						
		240	160	120	96	80	69	60
½	1	*6.2* **15**	*4.8* **12**	*4.1* **10**	*3.4* **9**	*2.9* **8**	*2.6* **7.5**	*2.4* **7**
¾	2	*13.2* **18**	*10.3* **14**	*8.6* **12**	*7.3* **11**	*6.2* **10**	*6.0* **9**	*5.5* **8.5**
1	2.3	*25.0* **22**	*19.2* **17**	*16.3* **15**	*14.4* **13**	*12.5* **12**	*12.0* **11**	*11.1* **10.5**
1¼	3.0	*52.8* **27**	*40.8* **21**	*34.8* **18**	*31.2* **16**	*27.8* **15**	*26.4* **14**	*24.0* **13**
1½	3.5	*79.2* **30**	*60.7* **23**	*51.2* **20**	*45.6* **18**	*40.8* **16**	*40.0* **15**	*36.0* **14**
2	4.0	*153.8* **36**	*120.0* **28**	*104.0* **24**	*93.5* **22**	*86.4* **20**	*81.5* **18**	*73.8* **17**
2½	6.0	*250.0* **41**	*192.0* **32**	*164.5* **28**	*149.0* **25**	*139.2* **22**	*135.8* **21**	*122.5* **19**
3	6.5	*444.0* **48**	*348.0* **37**	*294.0* **32**	*270.0* **29**	*254.0* **26**	*240.0* **24**	*223.0* **22**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

systems, it will be economical to operate with higher friction heads, and tables may be prepared similar to Tables 3 and 4, which are based on total friction heads of 6 and 18 ft, respectively.

Example 2. Design a direct return two-pipe forced circulation system for the layout shown in Fig. 5. For this system the length of the pipe line from the boiler to the highest radiator on the farthest riser and back to the boiler is about 250 ft. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so the total equivalent pipe length is about 300 ft.

Solution. The same pipe size tables may be used as those developed for the reversed return system of Fig. 3. Since this system is somewhat larger than that shown in Fig. 3, Table 3 which provides for a friction head of 6 ft may be used instead of Table 2 which provides for a friction head of only 2 ft.

Referring to the column for an equivalent total length of 300 ft for Sections *AB* and *KA*, each supplying 117.6 Mbh, it will be found that a 1½-in. pipe is too small and a 2-in. pipe is too large. Consequently, a 1½-in. pipe is selected for the flow line *AB*, and a 2-in. pipe for the return line *KA*. For Sections *BC* and *JK*, each supplying 88 Mbh, a 1½-in. pipe is only slightly too small and it is selected. The remaining pipe sizes are selected in a similar manner and recorded in Fig. 5. For a temperature drop of 10 F, 24.5 gpm of water must be circulated. The pump to select is one which has its highest efficiency when it is delivering 24.5 gpm against a 6-ft head.

TABLE 3. CAPACITIES OF PIPES IN *Mbh* (1000 BTU PER HOUR) AND **Velocities of Water in Pipes in Inches per Second** FOR FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 6 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F[a]

1	2	3	4	5	6	7	8
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit					
		200	300	400	600	800	1000
		Unit Friction Head, in Milinches					
		360	240	180	120	90	72
½	1	*7.4* **18**	*6.0* **15**	*5.0* **13**	*3.8* **10**	*3.4* **9**	*3.1* **7.5**
¾	2	*15.8* **22**	*12.7* **18**	*10.8* **16**	*8.4* **12**	*7.7* **11**	*6.7* **9**
1	2.5	*30.0* **27**	*24.0* **22**	*20.4* **19**	*15.8* **15**	*13.9* **13**	*12.5* **11**
1¼	3.3	*64.8* **33**	*52.5* **26**	*44.4* **23**	*33.6* **18**	*30.0* **16**	*26.8* **14**
1½	4.0	*96.0* **37**	*76.8* **31**	*64.8* **26**	*50.1* **20**	*44.7* **18**	*40.8* **15**
2	5.0	*192.0* **44**	*153.0* **36**	*130.0* **30**	*100.1* **24**	*90.0* **21**	*78.0* **18**
2½	6.0	*300.0* **50**	*244.0* **41**	*206.0* **35**	*161.0* **26**	*144.0* **24**	*130.0* **21**
3	7.5	*550.0* **58**	*436.0* **48**	*368.0* **42**	*287.0* **32**	*249.0* **27**	*228.0* **24**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

To secure a correct distribution of hot water among the several risers it is necessary, as previously stated, to introduce special resistances to balance the several risers, as follows:

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, *i.e.*, that they may operate under equal pressure heads, resistance must be added to the first riser equal to the friction head in the 80 ft of flow main from *B* to *F* plus that in the 80 ft of return main from *G* to *K*.

It will be noted from Table 3 that the unit friction head is about 240 milinches per foot. The total friction head in the flow and return mains between the first and fifth risers is therefore 160 × 240 or 38,400 milinches, or a little more than 3 ft, which must be supplied by additional resistance in the first riser to prevent its having an advantage over the fifth riser.

This resistance can be supplied by a calibrated and adjusted modulating valve or by an orifice resistor in a union. If the orifice resistor is to be used, its size may be selected from Table 5 as follows:

The lower part of the first flow riser supplies 28.8 Mbh. According to Table 3, it should be a 1-in. pipe and would have a velocity of 22 in. per second, if it were supplying 24 Mbh. Since it is supplying 28.8 Mbh, the velocity will be about 26 in. per second. From Table 5 it will be found that for a 1-in. pipe and a velocity of 24 in. per second, an 0.45-in. orifice will produce a loss of head of 37,000 milinches. For a velocity of 26 in. per second, the loss of head will be somewhat more, probably about 43,000 milinches; the

Table 4. Capacities of Pipes in *Mbh* (1000 Btu per Hour) and **Velocities of Water in Pipes in Inches per Second** for Forced Circulation Systems with a Total Friction Head of 18 ft and for a Maximum Temperature Drop of 10 F[a]

1	2	3	4	5	6	7
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit				
		200	400	600	800	1000
		Unit Friction Head, in Milinches				
		1080	540	360	270	216
½	1.0	*12.7* **32**	*8.6* **23**	*7.2* **18**	*6.2* **15**	*5.5* **13**
¾	2.0	*27.5* **40**	*18.7* **28**	*15.1* **22**	*13.7* **19**	*11.5* **17**
1	2.5	*55.0* **48**	*36.8* **34**	*30.0* **27**	*26.4* **23**	*22.6* **20**
1¼	3.0	*122.0* **59**	*81.5* **42**	*66.0* **33**	*58.3* **28**	*50.5* **25**
1½	4.0	*182.0* **66**	*122.0* **46**	*98.2* **37**	*86.2* **31**	*74.2* **27**
2	5.0	*371.0* **80**	*252.0* **56**	*201.0* **45**	*180.0* **38**	*151.0* **33**
2½	7.0	*598.0* **91**	*407.0* **65**	*323.0* **51**	*287.0* **43**	*240.0* **38**
3	9.0	*1110.0* **107**	*790.0* **76**	*598.0* **60**	*527.0* **51**	*443.0* **44**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

difference between it and the required resistance will be about 10 per cent, which is permissible, and the 0.45-in. orifice is selected.

The sizes of the orifice resistors for the second, third, and fourth risers are selected in a similar manner and found to be 0.45 in., 0.50 in., and 0.55 in., respectively.

If the design of the system of Fig. 5 is to be extremely refined, the gravity pressure heads produced by the risers should be taken into consideration. With water at 220 F and 210 F, respectively, in the risers, the gravity head is 50 milinches per foot of water column or 25 milinches per foot of flow and return pipe. The pump pressure head in this case is 240 milinches per foot of pipe, and the gravity head, being only one tenth as large as the pump head, may be neglected without serious error. This is generally done.

Temperatures of 220 F and 210 F would be used only during the coldest

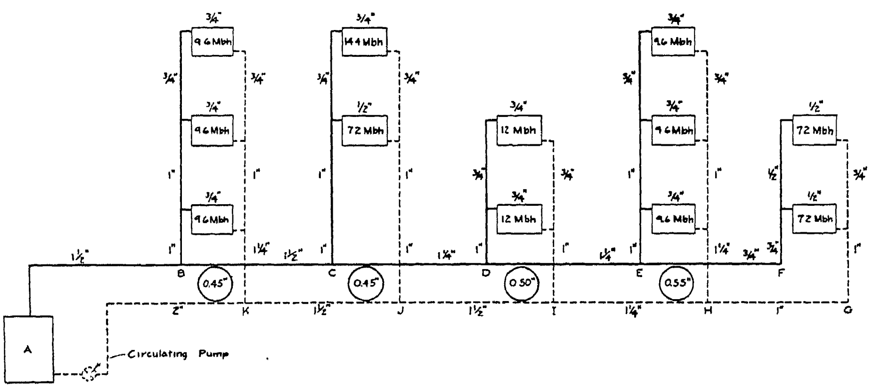


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

weather for which the system is designed. At other times the temperatures would be lower, the temperature drop smaller, and the gravity heads smaller. The pump pressure head remains constant throughout the season if the pump is operated at a constant speed and, consequently, the gravity head is generally less than one-tenth of the pump head.

Effect of Variations in Pipe Sizes

The pipe sizes for the several parts of the system selected from the tables are only approximately correct but the resulting error should be negligible as may be seen from the following study. Assume, as an extreme case, that the error in pipe size is so large that the water flows twice as fast through one of the radiators as through the others. This would make the friction head through this radiator almost four times as large as those through the other radiators. The result would be that the water, in flowing through the radiator, would cool 5 F instead of 10 F. The mean water temperature in the radiator would then be 217½ F instead of 215 F, and the mean temperature difference, water to air, would be 147½ F instead of 145 F. The heat dissipated by the radiator would therefore be about 2 per cent more than calculated. It is evident that this difference in heat dissipation is smaller than the difference between

TABLE 5. FRICTION HEADS (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

Diameter of Orifices (Inches)	Velocity of Water in Pipe in Inches per Second									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

the calculated heat losses and the actual heat losses, and also smaller than the average difference between the calculated radiator sizes and the nearest stock sizes selected.

GRAVITY CIRCULATION

For gravity circulation, the one-pipe system shown in Fig. 6 and the two-pipe direct return system shown in Fig. 7 are probably in most common use.

The one-pipe system has the disadvantage that the radiator nearest the

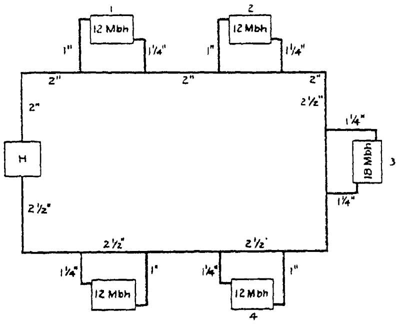


FIG. 6. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

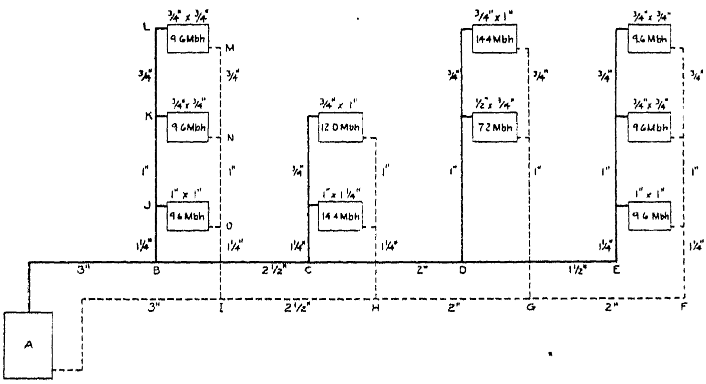


FIG. 7. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

boiler is the only one which receives water at approximately the temperature at which it leaves the boiler. All other radiators receive cooler water and must be proportionally increased in size, so the total heating surface in the system is considerably larger than that in a corresponding two-pipe system.

The pipe sizes in gravity circulation systems may be varied. As the pipe sizes are decreased, the temperature drop through the radiators, which produces circulation, is increased and it becomes necessary to increase the temperature of the water leaving the boiler so that the mean temperature in the radiator remains constant. For example, Fig. 8 shows

diagrammatically an elementary heating system which will function with either 1½-in. or 1-in. pipe. The radiator is required to deliver 27 Mbh, and the circuit consists of 30 ft of pipe and 20 elbow equivalents.

If 1½-in. pipe is used, the system will operate correctly if the water temperatures in the flow and return risers are 200 F and 180 F, respectively. The mean water temperature in the radiators will then be 190 F and, if the radiator is located in air having a temperature of 70 F, the size of the radiator must be sufficient to deliver 27 Mbh under these conditions.

If 1-in. pipe is used, the system will function correctly with water temperatures in the flow and return risers of 210 F and 170 F, or of 200 F and 160 F. In the first case, the mean water temperature is again 190 F and the same size radiator may be used as with the 1½-in. pipe, but the temperature of the water leaving the boiler must be raised from 200 F to 210 F. In the second case, the temperature of the water leaving the boiler is the same as for the 1½-in. pipe, but the mean water temperature

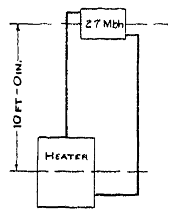


FIG. 8. AN ELEMENTARY SYSTEM

in the radiator is lowered from 190 F to 180 F, and theoretically the size of the radiator should be increased about 12½ per cent to deliver the required 27 Mbh (See Table 3, Chapter 6, 1933 GUIDE).

This indicates the extent to which pipe sizes and radiator sizes may be decreased by increasing the temperatures of the water in the boiler, as is possible in closed systems and in open systems in which the open expansion tank is located sufficiently high to secure a pressure in the boiler equal to that existing in the boiler of the closed system.

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 6. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 6, Column 2, a 2-in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 6 may be used to determine the size of the mains. Note from Column 8, for a 200-ft length, that a 2-in. main will supply 48 Mbh and a 2½-in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2-in. pipe is too small and a 2½-in. pipe too large. The solution is to use some 2-in. and some 2½-in. pipe. Since the 2½-in. is nearer the correct size than the 2-in., select 2-in. pipe for the first 50 or 60 ft out of the boiler and 2½-in. for the remaining pipe back to the boiler.

Tables 7 and 8 may be used to design the radiator risers and connections. According to Table 7, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 8, both radiator tappings should be 1 in. To simplify the construction, select 1-in. flow risers with 1-in. riser branches and 1-in. radiator tappings. Also select 1¼-in. return risers with 1¼-in. riser branches, and 1¼-in. radiator tappings. Similarly, for 18 Mbh, select 1¼-in. flow and return risers and riser branches, and 1¼-in. radiator tappings.

TABLE 6. CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F, WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[a])	Equivalent Total Length of Pipe in Feet in Longest Circuit								
		75	100	125	150	175	200	250	300	350
		Unit Friction Head, in Milinches								
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7
1½	3.0	*43.0*	*37.5*	*33.0*	*30.0*	*27.0*	*25.0*	*22.2*	*20.2*	*18.7*
2	4.0	*83.0*	*72.0*	*63.0*	*57.0*	*51.0*	*48.0*	*42.0*	*38.0*	*35.0*
2½	4.5	*140.0*	*115.0*	*100.0*	*90.0*	*81.5*	*75.4*	*67.2*	*61.0*	*56.0*
3	5.0	*234.0*	*204.0*	*175.5*	*160.0*	*143.0*	*133.0*	*110.0*	*107.5*	*100.0*
3½	5.5	*347.0*	*300.0*	*260.0*	*236.0*	*214.0*	*200.0*	*177.0*	*160.0*	*146.0*
4	6.0	*490.0*	*422.0*	*370.0*	*334.0*	*297.0*	*278.0*	*248.0*	*223.0*	*205.0*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 6, in which the total temperature drop is to be 35 F and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the

TABLE 7. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND **Velocities of Water in Pipes in Inches Per Second** FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

Pipe Size (Inches)		Equivalent Length of Pipe (Feet[c])	1st Floor[b]			2nd Floor	3rd and 4th Floors
				Vel. (In. per Sec.)[d]			
Flow	Return		*Mbh*	Flow	Return	*Mbh*	*Mbh*
½	½	1.0				*5*	*6.2*
½	¾					*6.4*	*8.0*
¾	¾	1.5	*9*	**2.3**	**2.3**	*10.1*	*14.0*
¾	1		*12*	**3.2**	**2.0**	*12.8*	*17.1*
1	1	2.0	*18*	**2.5**	**2.5**	*20*	*26.0*
1	1¼		*21*	**3.0**	**2.0**	*25.2*	*34*
1¼	1¼	3.0	*26*	**3.0**	**3.0**	*43*	*55*
1¼	1½		*34*	**4.0**	**2.5**		
1½	1½	3.5	*48*	**3.0**	**3.0**		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.

[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.

[c]Approximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.

[d]Velocities apply to the riser branches.

water in the seventh radiator will be 170 F, and, according to Table 3, Chapter 6, of the 1933 Guide, the heat dissipation of these two radiators will be to each other as 868 is to 617, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in., the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in. of water, or about one-fortieth of the pressure head produced by the circulating pump selected for the system of Fig. 3.

Table 8. Maximum Capacities of Radiator Connections in *Mbh*, for One-Pipe and for Two-Pipe Direct Return Gravity Circulation Systems with a Temperature Drop of 35 F Through Each Radiator

Pipe Size		Equivalent Length of Pipe (Feet[a])	1st Floor	2nd, 3rd, and 4th Floors
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	*4.1*	*5.9*
½	¾		*5.2*	*7.5*
¾	¾	1.5	*7.0*	*10.5*
¾	1		*9.1*	*13.0*
1	1	2.0	*12.5*	*17.8*
1	1¼		*17.5*	*23.2*
1¼	1¼	3.0	*23.3*	*33.2*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

Table 6 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3-in. pipe is too large and a 2½-in. pipe is too small; hence, select 2½ in. for Section *AB* and 3 in. for Section *IA*. For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½-in. pipe is almost exactly the correct size and is selected for both sections.

For the forced circulation system of Fig. 5, the pressure head produced by the circulating pump is used to force the water through the mains and also through the risers. Gravity circulation systems have two distinct pressure heads. One is produced by the difference in weight of the water in the flow and return risers adjacent to the boiler, and is the boiler pressure head, which in this case is 0.6 in. The other pressure head is produced by the difference in weight of the water in the flow and return risers adjacent to the radiators, and is the radiator pressure head. If the temperature drop through the radiators is about 35 F, and if the story heights of the building are 9 ft and the distance from the center of the first floor radiator to the average level of the main is 3 ft, the radiator pressure head of the first floor radiator is about 450 milinches and the pressure heads of the radiators on the upper floor are 1350 milinches greater than those on the next lower floors.

Tables 6 and 7 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 7 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh. According to the table, 1¼-in. flow and return risers will supply 26.0 Mbh; if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and 1¼-in. flow and return risers are selected. However, it must be remembered that the riser

branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1-in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a ½-in. flow and a ¾-in. return riser are used, the capacity will be 8.0 Mbh; if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾-in. pipe is selected for both risers.

To design the radiator connections, use Table 8 and note that for the first floor radiator connections the capacity of a ¾-in. flow and 1-in. return is 9.1 Mbh, and that of a 1-in. flow and a 1-in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1-in. flow and return connection is selected. For the two upper floors, the capacity of a ¾-in. flow and return connection is 10.5 Mbh, and that size is used.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 7 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (See Table 6, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess of that available for the fourth set. The velocity in the riser branch is 3 in. per second (See Table 7) and, therefore, according to Table 5, an 0.65-in. orifice in a 1¼-in. union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70-in. orifice in a 1¼-in. union will provide a resistance of 285 milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60-in. orifice in a 1-in. union will provide sufficient resistance.

MECHANICAL CIRCULATION

Circulating pumps for hot water systems may be used to provide the motive head for forced circulation systems as already described, or to improve the operation of gravity-designed systems. Small specially-designed centrifugal pumps installed on a by-pass with the necessary gate or check valves near the point where the return main enters the heater may be employed. Specially-designed, electrically-driven, propeller-type circulating pumps or units may also be employed. The latter are usually installed directly in the return main and are available for all commercial pipe sizes used for hot water heating. The motor switch may be under manual control, automatic control using thermostatic elements, or tied in with the oil or gas burner switch which starts and stops the burner. For large capacities these units may be installed in multiple.

For exceptionally large installations such as central heating plants, circulating pumps of the centrifugal single stage type, having an average operating efficiency of 70 per cent against heads up to 125 ft, are sometimes used. It is generally advisable to install the pumps in duplicate to provide for contingencies and to insure continuous operation. In such cases each pump may be made equal to two-thirds of the maximum capacity required.

EXPANSION TANKS

When water at ordinary temperatures is heated or cooled, its volume is increased or decreased. This variation in the volume of the water in a heating system is generally provided for by means of an expansion tank into which the water can flow from the system during the heating-up periods and from which it can flow back into the system during the cooling-down periods.

The expansion tank may be open or closed. In an open expansion tank (Fig. 9), the water is subjected to atmospheric pressure and can expand freely without a material increase in pressure. In a closed expansion tank (Fig. 10), the water is subjected to the pressure of the compressed air

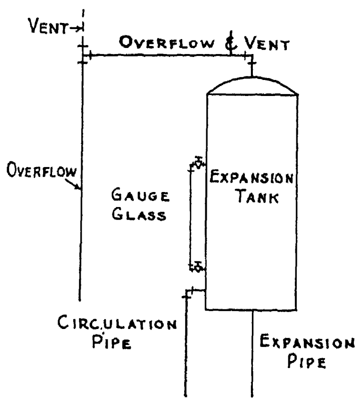


FIG. 9. AN OPEN EXPANSION TANK

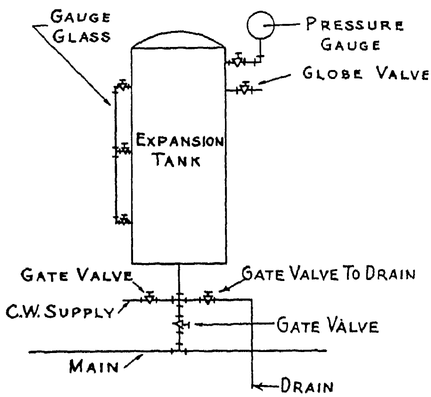


FIG. 10. A CLOSED EXPANSION TANK

within the tank, and as the water expands, the volume of the air in the tank is decreased and its pressure increased.

The open expansion tank must be placed at a sufficient elevation above the highest radiator to prevent boiling when the water in that radiator is at the highest temperature to which it is to be heated. For example, if the water is to be heated to 225 F on extremely cold days, the absolute pressure on the water in the highest radiator must be at least 19 lb per square inch. This pressure will be secured if the open expansion tank is located 15 ft above the highest radiator. If a closed expansion tank is used and is located 30 ft below the highest radiator, an absolute pressure of about 32 lb per square inch must be maintained in the expansion tank if the water in the highest radiator is to be heated to 225 F without danger of boiling.

The type of expansion tank used in a heating system, whether open or closed, has no influence on the operation of the system. The only function performed by the expansion tank is to provide for the variation in the volume of the water in the system, and at the same time to maintain a sufficient pressure in the system to prevent boiling when the water is at the highest temperature for which the system is designed. The use of an expansion tank may be dispensed with when the heating system is allowed to *float* on the water system, *i.e.*, when the connection between

the heating system and the water system is kept open so that the water system replaces the expansion tank.

The capacity of the expansion tank should be at least twice the increase in volume produced when the water in the system is heated from its normal to its maximum temperature. When 25 gal of water are heated from 40 F to 200 F, the volume of water increases to 26 gal. A safe rule, therefore, is to make the water capacity of the expansion tank equal to 10 per cent of the capacity of the heating system.

In a forced circulation system, the expansion tank should be connected to the return main near the circulating pump. In a gravity circulation system, the expansion tank should be connected to the flow riser so that air liberated from the water in the boiler may escape through the expansion tank, except where it is desired to maintain a temperature higher than 212 F, in which case the connection should be in the return main to prevent possible boiling in the expansion tank.

The expansion tank should be protected so that the water in the tank or in the connecting pipe lines cannot freeze. If such water should freeze and the water in the system be heated to cause further expansion, the resulting force will burst the boiler or some other portion of the system.

INSTALLATION DETAILS

The detailed installation of the pipe system should be governed by four fundamental rules:

1. All piping must be pitched either up or down so that all gases which are liberated from the water can move freely to a vented section of the system. Whenever practicable, the pipe line should be pitched so that gases flowing to a vent will flow in the same direction as the water. When a pipe system cannot be installed without creating *air pockets*, that is, sections in the system from which liberated gases cannot escape, such sections must be provided with automatic air relief valves or with air valves which may be operated manually when necessary.

2. All piping must be arranged so that the entire system can be drained, either to permit alterations or repairs, or to prevent freezing if the system is not to be operated during a cold period.

It is well to install a gate valve and union in every riser near the main to permit the draining of individual risers without draining the entire system. It is also well, in large installations, to divide the system into branches and to provide each branch with unions and valves so that any one branch can be drained without disturbing the remaining ones.

The dividing of large heating systems into branches or zones and providing each zone with individual valves has the further advantage of permitting a varying temperature control. For example, if a building is equipped with a forced circulating system and if the south rooms are on one branch of the main and the north rooms are on a separate branch, the valves may be set so that the water will circulate through the north branch with a temperature drop of, say, 10 F, and through the south branch with a temperature drop of, say, 20 F, thus delivering less heat to the south rooms than to the north rooms. This arrangement is especially valuable when the regulating valves are controlled thermostatically by the temperatures in the two zones, because no matter how accurately the heating system may have been designed, the heat demand of any group of rooms varies with sunshine and with wind velocity, and these intermittent variations can be provided for only by the individual control made possible by changing the valve settings controlling the heat supplied to particular groups of rooms.

3. All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

4. The pipe system must be installed so that each circuit has its correct friction head. To bring this about, it is necessary in some cases to minimize the friction, *i.e.*, to make the pipe line as short as possible and to provide as few fittings as possible; and in other cases it is necessary to increase the length of the pipe and the number of fittings so that, for every circuit, the friction head will be equal to the available pressure head.

The connections from the boiler to the mains should be short and direct, to reduce the friction head. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft. The flow main should always be covered; the return main should be covered except where it is to provide the heating surface for the basement.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return

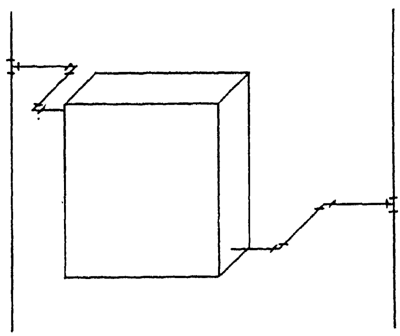


FIG. 11. METHOD OF CONNECTING RADIATOR TO ALLOW FOR EXPANSION OF PIPE

main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side, but in most cases both connections should be at an angle of 45 deg. This method shortens the lines and substitutes 45-deg ells for 90-deg ells.

Preferably, connection of the flow riser to a radiator should be to the upper tapping, and connection of the return riser to a radiator should be to the lower tapping. When hot water enters at the top of a radiator it will distribute itself along the entire length of the radiator, and as it cools it will settle gradually to the bottom; the cool water may then be taken out of the radiator at either end.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation and low velocities it makes little difference whether the water leaves at the end at which it enters or at the opposite end.

The connections of the risers to the radiators should be such that provision is made for the vertical expansion of the risers. This can be accomplished as indicated in Fig. 11 by using one tee and two ells for each connection. These connections should be pitched upward or downward, whichever may be necessary to prevent the formation of air pockets and to permit draining.

PROBLEMS IN PRACTICE

1 ● What causes the circulation of water in hot water heating systems?

In gravity systems, circulation is caused by the difference between the weight of the cool water in the return riser and that of the hot water in the flow riser.

In forced circulation systems, circulation is produced primarily by a pump, and secondarily by the difference in the weights of the water in the return and flow risers. However, the secondary effect is so small when compared with that of the circulating pump that it may be neglected in most cases.

2 ● What tends to prevent or to retard the circulation of water in hot water heating systems?

In both gravity flow and forced circulation systems, the friction which must be overcome when the water is flowing through pipes, fittings, valves, heaters, and radiators tends to prevent or retard circulation. For a given pipe the friction varies approximately as the 1.7 power of the velocity, and for given fittings, valves, heaters, and radiators, the friction varies approximately as the square of the velocity. It is therefore sufficiently accurate to express the friction in fittings, valves, heaters, and radiators in terms of the friction in one standard elbow, as shown in Table 1.

3 ● In the elementary heating system, Fig. 8, what is the pressure head maintaining the circulation if the water in the return riser is at 180 F and that in the flow riser is at 200 F?

It is found, from Table 8, Chapter 1, that 180 F water weighs 60.61 lb per cu ft and 200 F water weighs 60.13 lb per cu ft. The pressure head is independent of the size of the pipe. If the two risers were each 1 ft square, the water in the flow riser would weigh 601.3 lb and that in the return riser would weigh 606.1 lb. Thus the water in the return riser would weigh 4.8 lb more than that in the flow riser. Consequently, the resulting pressure head is 4.8 lb per square foot.

Pressure heads are generally expressed in feet, or inches, or milinches of water of a given temperature. In this case we are dealing with water at both 180 F and 200 F, so the pressure head is expressed in terms of 190 F water. Such water weighs 60.39 lb per cu ft, and to secure a pressure of 4.8 lb per square foot, it is necessary to have a column of water having a weight of 4.8 divided by 60.39 = 0.0795 ft, or 0.9540 in., or 954 milinches. This is the pressure head which maintains the circulation.

4 ● In the elementary system of Question 3, if the radiator dissipates 14,000 Btu per hour, what is the velocity of the water in the pipe line, if the pipes are 1 in. in diameter? What, if they are ¾ in. in diameter?

Since the temperature drop through the radiator is from 200 F to 180 F or 20 F, every pound of water flowing through the radiators delivers 20 Btu; consequently, 14,000 divided by 20 = 700 lb of water, or for 190 F water, 700 divided by 60.39 = 11.59 cu ft of water must flow through the radiator and through the pipe lines every hour.

The interior area of a 1-in. pipe is 0.864 sq in. The velocity in the 1-in. pipe is 11.59 divided by 0.864 and multiplied by 144 = 1932 ft per hour or 6.44 in. per second.

For ¾-in. pipe, the interior area is 0.533, and the velocity is 6.44 multiplied by 0.864 and divided by 533 = 10.44 in. per second.

5 ● If, in the elementary heating system of Question 3, a 1-in. pipe line is used, what would be the friction head?

If the radiator is connected as shown in Fig. 11, with the heater connected to provide freedom of expansion, the heating circuit may be assumed to consist of a heater, 25 ft of pipe, 8 elbows, 1 radiator valve, and 1 radiator. From Table 1 it appears that the heater and radiator are equivalent, in friction, to 6 elbows; hence, the circuit may be placed equal to 25 ft of pipe and 14 elbows.

From the diagram of Fig. 4 it appears that the friction head for a 1-in. pipe and a velocity

of 6.44 in. per second is about 25 milinches per foot. For 25 ft of pipe, the friction head will be 625 milinches.

It appears from Table 1 that the friction head in one elbow is $\frac{v^2}{2g}$, or in this case 0.54 multiplied by 0.54 and divided by 64.4 = 0.0045 ft or 54 milinches. Hence, for the 14 elbows the friction is 756 milinches. For the entire circuit, the friction head is the sum of the 625 milinches of the pipe plus the 756 milinches of the elbows, or 1381 milinches which equal 1.381 in.

6 ● If the elementary heating system of Question 3 is installed with a 1-in. pipe line, how will it function?

It is found from the answer to Question 3 that the pressure head is 954 milinches and from the answer to Question 5 that the friction head is 1381 milinches when the water is flowing with such velocity that the specified 14,000 Btu will be delivered with a 20 F temperature drop through the radiators. Since the pressure head is smaller than the friction head, the system will not function as planned for the water will flow through the system more slowly and remain in the radiator longer. The temperature drop through the radiator will be more than 20 F, and the difference in the weight of the water in the return and flow risers will be greater than that intended. The final result will be that the pressure head will become equal to the friction head at a value somewhere between 954 and 1381 milinches. Since the average water temperature in the radiator will be less than 190 F, the radiator should be larger than the size given in Question 4.

7 ● Should a hot water heating system be designed to embody small pipes or large pipes?

As pipe sizes in gravity circulation heating are reduced, the friction head is increased and it is necessary to increase the temperature drop through radiators; this lowers the average temperature of the water in the radiators and necessitates an increase in the size of the radiators, so whereas the cost of the pipe in a system is reduced, the cost of the radiators is increased. For each installation there is a definite pipe size which entails maximum economy.

As pipe sizes in forced circulation systems are reduced, friction heads are increased so a circulating pump of greater size or capacity is required. Thus, by decreasing the size of the piping, both the first cost of the circulating pump and the cost of its operation are increased. There is a definite pipe size for every installation which is most economical. For each installation of both types of systems there is a definite pipe size entailing maximum economy which can be determined by a series of comparative calculations.

8 ● What should be the size of the radiators for the elementary heating system of Question 3 in which the water enters the radiator with a temperature of 200 F and leaves with a temperature of 180 F? The average temperature of the water in the radiator is, approximately, 190 F.

If test results are available for the particular radiators to be used, and for the temperatures named, the size of the radiators should be selected from them. If no such test results are to be had, but if test results are available for the type of radiator to be used when it is supplied with 215 F steam and placed in a 70 F room, the required size may be determined by the following ratio: The required size is to the corresponding steam radiator size as $(215 - 70)^{1.3}$ is to $(190 - 70)^{1.3}$. This ratio works out to 1.28. Hence, the radiators should be 28 per cent larger under the conditions prescribed than are corresponding radiators under standard conditions. It is immaterial whether a radiator is filled with steam or with water, as long as the average temperature of its outer surface is the same in both cases.

Chapter 34

PIPE, FITTINGS, WELDING

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used for heating and ventilating work at the present time is of wrought steel. The material used for steel pipe is a mild steel made by the acid-bessemer, the open-hearth, or the electric-furnace process. Ordinary wrought-steel pipe is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. The former is known as *welded pipe,* the latter as *seamless pipe.*

Many types of welded pipe are available, although the smaller sizes most frequently used in heating and ventilating work are made by the lap-weld or butt-weld process. While the lap-weld process produces a better weld than the butt type, lap-weld pipe is seldom manufactured in nominal pipe sizes less than 2 in. Seamless pipe can be obtained in the small sizes at a somewhat higher cost.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat greater strength which permits use of a thinner wall and, in the small sizes, its freedom from the occasional tendency of welded pipe to split at the weld when bent.

Wrought-Iron Pipe. Wrought-iron pipe is considered to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher first cost can be justified on the basis of longer life expectancy. Wrought-iron pipe may be identified by the spiral line marked into each length, either knurled into the metal or painted on it in red or other bright color.

Otherwise, there is little difference in the appearance of wrought-iron and steel pipe, although microscopic examination of polished and etched specimens will readily disclose the difference.

Cast Ferrous Pipe. There are now available several types of cast ferrous-metal pipe made of a good grade of cast-iron with or without additions of nickel, chromium, or other alloy. This pipe is available in sizes from 1½ in. to 6 in., and in standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra-strong wrought pipe. Cast ferrous pipe may be obtained coupled, bevelled for welding, or with ends plain or grooved for the several types of couplings. It is easily cut and threaded as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practicable for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper or other alloying element, such as molybdenum or manganese, has been claimed to possess more resistance to corrosion than plain steel pipe and it is advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and brass pipe have always been used in heating, ventilating, and water supply installations, but the cost with standard dimensions for threaded connections has been high. The recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin-wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought-iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double-extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall thicknesses listed. In sizes 14 in. and upward, pipe is designated by its outside diameter (O.D.) and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by the *standard-weight* and *extra-strong* pipe,

demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Even in heating installations, the erection of piping by welding was deemed to warrant the use of pipe lighter than standard weight. For these reasons, a *Sectional Committee on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing* functioning under the procedure of the *American Standards Association* was appointed to standardize the dimensions and materials of pipe.

The proposed pipe standard recommended by that sectional committee has set up several schedules of pipe including standard-weight and extra-strong thicknesses which are now included in Schedules 40 and 60, respectively. The schedules approved by the Sectional Committee are given in Tables 1 and 3 and the corresponding weights in Tables 2 and 4.

Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent of the total number of lengths ordered may be *jointers* which are two pieces coupled together. Extra-strong pipe is generally furnished with plain ends in random lengths of 12 to 22 ft, although 5 per cent may be in lengths of 6 to 12 ft.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature one hundred degrees or more above room temperature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in temperature is termed the *coefficient of linear expansion* of that material, or commonly, the *coefficient of expansion.* This coefficient varies with the material.

The linear expansion of cast iron, steel, wrought-iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 5.

The elongation values in Table 5 were computed from the following formula:

$$L_t = L_o \left[1 + a\left(\frac{t-32}{1000}\right) + b\left(\frac{t-32}{1000}\right)^2\right] \tag{1}$$

where

L_t = length at temperature t degrees Fahrenheit, feet.
L_o = length at 32 F, feet.
t = final temperature, degrees Fahrenheit.
a and b are constants as follows:

METAL	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought-Iron	0.006503	0.001622
Copper	0.009278	0.001244

The three methods by which the elongation due to thermal expansion may be taken care of are:

1. Expansion joints.
2. Swivel joints.
3. Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking

Table 1. Dimensions of Welded and Seamless Steel Pipe

Nominal Pipe Size	Outside Diam.	Nominal Wall Thicknesses for Schedule Numbers									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
1/8	0.405	...	...	...	0.068*	...	0.095*	...	...	...	...
1/4	0.540	...	...	...	0.088*	...	0.119*	...	...	...	...
3/8	0.675	...	...	...	0.091*	...	0.126*	...	...	...	...
1/2	0.840	...	...	...	0.109*	...	0.147*	...	...	...	0.187
3/4	1.050	...	...	...	0.113*	...	0.154*	...	...	...	0.218
1	1.315	...	...	...	0.133*	...	0.179*	...	...	...	0.250
1 1/4	1.660	...	...	...	0.140*	...	0.191*	...	...	...	0.250
1 1/2	1.900	...	...	...	0.145*	...	0.200*	...	...	...	0.281
2	2.375	...	...	...	0.154*	...	0.218*	...	...	...	0.343
2 1/2	2.875	...	...	...	0.203*	...	0.276*	...	...	...	0.375
3	3.500	...	...	...	0.216*	...	0.300*	...	...	...	0.437
3 1/2	4.000	...	...	...	0.226*	...	0.318*	...	...	...	...
4	4.500	...	...	...	0.237*	...	0.337*	...	0.437	...	0.531
5	5.563	...	...	...	0.258*	...	0.375*	...	0.500	...	0.625
6	6.625	...	...	...	0.280*	...	0.432*	...	0.562	...	0.718
8	8.625	...	0.250	0.277*	0.322*	0.406	0.500*	0.593	0.718	0.812	0.906
10	10.75	...	0.250	0.307*	0.365*	0.500*	0.593	0.718	0.843	1.000	1.125
12	12.75	...	0.250	0.330*	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O. D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O. D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O. D.	30.0	0.312	0.500	0.625	...	...	...	...	...	...	...

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under nominal thicknesses.

*Thicknesses marked with asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since continued turning in the threaded joint may in time result in a leak, par-

ticularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means

TABLE 2. NOMINAL WEIGHTS OF WELDED AND SEAMLESS STEEL PIPE

NOMINAL PIPE SIZE INCHES	SCHED. 10 PLAIN ENDS	SCHED. 20 PLAIN ENDS	SCHEDULE 30		SCHEDULE 40		SCHED. 60 PLAIN ENDS	SCHED. 80 PLAIN ENDS	SCHED. 100 PLAIN ENDS	SCHED. 120 PLAIN ENDS	SCHED. 140 PLAIN ENDS	SCHED. 160 PLAIN ENDS
			Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings						
1/8					0.25*	0.25*		0.32*				
1/4					0.43*	0.43*		0.54*				
3/8					0.57*	0.57*		0.74*				
1/2					0.86*	0.86*		1.09*				1.31
3/4					1.14*	1.14*		1.48*				1.94
1					1.68*	1.69*		2.18*				2.85
1 1/4					2.28*	2.29*		3.00*				3.77
1 1/2					2.72*	2.74*		3.64*				4.86
2					3.66*	3.68*		5.03*				7.45
2 1/2					5.80*	5.82*		7.67*				10.0
3					7.58*	7.62*		10.3*				14.3
3 1/2					9.11*	9.21*		12.5*				
4					10.8*	10.9*		15.0*		19.0		22.6
5					14.7*	14.9*		20.8*		27.1		33.0
6					19.0*	19.2*		28.6*		36.4		45.3
8		22.4	24.7*	25.0*	28.6*	28.8*	35.7	43.4*	50.9	60.7	67.8	74.7
10		28.1	34.3*	35.0*	40.5*	41.2*	54.8*	64.4	77.0	89.2	105.0	116.0
12		33.4	43.8*	45.0*	53.6	55.0	73.2	88.6	108.0	126.0	140.0	161.0
14 O. D.	36.8	45.7	54.6		63.3		85.0	107.0	131.0	147.0	171.0	190.0
16 O. D.	42.1	52.3	62.6		82.8		108.0	137.0	165.0	193.0	224.0	241.0
18 O. D.	47.4	59.0	82.0		105.0		133.0	171.0	208.0	239.0	275.0	304.0
20 O. D.	52.8	78.6	105.0		123.0		167.0	209.0	251.0	297.0	342.0	374.0
24 O. D.	63.5	94.7	141.0		171.0		231.0	297.0	361.0	416.0	484.0	536.0
30 O. D.	99.0	158.0	197.0									

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*The weights marked with asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists: those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 3, Chapter 32, shows several types of expansion bends commonly

[1]Piping Handbook, by Walker and Croker, and A Manual for the Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, and E. T. Cope, published by *The Detroit Edison Company*.

used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from the following formula:

$$L = 6.16 \sqrt{D \Delta} \tag{2}$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. It is

TABLE 3. DIMENSIONS OF WELDED WROUGHT-IRON PIPE

Nominal Pipe Size	Outside Diameter	Nominal Wall Thicknesses for Schedule Numbers					
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80
1/8	0.405	....	....	....	0.070*	....	0.098*
1/4	0.540	....	....	....	0.090*	....	0.122*
3/8	0.675	....	....	....	0.093*	....	0.129*
1/2	0.840	....	....	....	0.111*	....	0.151*
3/4	1.050	....	....	....	0.115*	....	0.157*
1	1.315	....	....	....	0.136*	....	0.183*
1 1/4	1.660	....	....	....	0.143*	....	0.195*
1 1/2	1.900	....	....	....	0.148*	....	0.204*
2	2.375	....	....	....	0.158*	....	0.223*
2 1/2	2.875	....	....	....	0.208*	....	0.282*
3	3.5	....	....	....	0.221*	....	0.306*
3 1/2	4.0	....	....	....	0.231*	....	0.325*
4	4.5	....	....	....	0.242*	....	0.344*
5	5.563	....	....	....	0.263*	....	0.383*
6	6.625	....	....	....	0.286*	....	0.441*
8	8.625	....	....	0.283*	0.329*	....	0.510*
10	10.75	....	....	0.313*	0.372*	0.510*	0.606
12	12.75	....	....	0.336*	0.414	0.574	0.702
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.625	0.750
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.687	....
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.750	....
20 O. D.	20.0	....	0.375	0.500	0.562	....	....

All dimensions are given in inches.
The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under the nominal thickness.
*Thicknesses marked with an asterisk in Schedules 30 and 40 are identical with thicknesses for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for *extra-strong* pipe in former lists.
The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

It is especially necessary with light-weight radiators so to anchor the piping and so to give it freedom for expansion that no strain therefrom shall be allowed to distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

Table 4. Nominal Weights of Welded Wrought-Iron Pipe

Nominal Pipe Size (Inches)	Sched. 10	Sched. 20	Schedule 30		Schedule 40		Schedule 60	Schedule 80
	Plain Ends	Plain Ends	Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings	Plain Ends	Plain Ends
1/8	....	....	....	....	0.25*	0.25*	....	0.32*
1/4	....	....	....	....	0.43*	0.43*	....	0.54*
3/8	....	....	....	....	0.57*	0.57*	....	0.74*
1/2	....	....	....	....	0.86*	0.86*	....	1.09*
3/4	....	....	....	....	1.14*	1.14*	....	1.48*
1	....	....	....	....	1.68*	1.69*	....	2.18*
1 1/4	....	....	....	....	2.28*	2.29*	....	3.00*
1 1/2	....	....	....	....	2.72*	2.74*	....	3.64*
2	....	....	....	....	3.66*	3.68*	....	5.03*
2 1/2	....	....	....	....	5.80*	5.82*	....	7.67*
3	....	....	....	....	7.58*	7.62*	....	10.3*
3 1/2	....	....	....	....	9.11*	9.21*	....	12.5*
4	....	....	....	....	10.8*	10.9*	....	15.0*
5	....	....	....	....	14.7*	14.9*	....	20.8*
6	....	....	....	....	19.0*	19.2*	....	28.6*
8	....	....	24.7*	25.0*	28.6*	28.8*	....	43.4*
10	....	....	34.3*	35.0*	40.5*	41.2*	54.8*	54.4
12	....	....	43.8*	45.0*	53.6	55.0	73.2	88.6
14 O. D.	36.0	44.8	53.6	....	62.2	....	87.6	104.0
16 O. D.	41.3	51.4	61.4	....	81.2	....	111.0	....
18 O. D.	46.5	57.9	80.5	....	103.0	....	136.0	....
20 O. D.	....	77.0	103.0	....	115.0	....	....	....

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*Weights marked with an asterisk in Schedules 30 and 40 are identical with weights for *standard-weight* pipe in former lists; those in Schedules 60 and 80 are identical with weights for *extra-strong* pipe in former lists.

The Schedule Numbers indicate approximate values of the expression 1000 x P/S.

HANGERS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or

supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel

TABLE 5. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39	....	60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45	....	200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From Piping Handbook, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support when expansion takes place, or the bottom support should be designed to withstand the entire load.

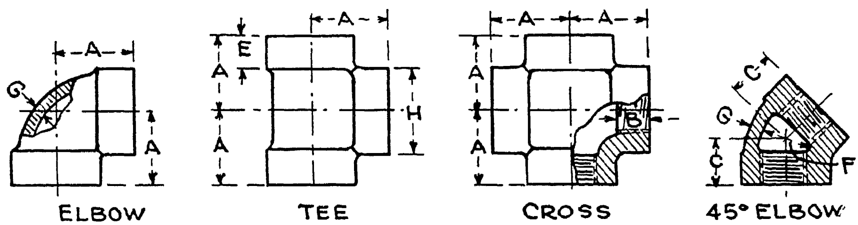
FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being

generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this malleable iron, steel and steel alloys are also used, as well as various

TABLE 6. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45 DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON SCREWED FITTINGS



	A	C	B	E	F		G	H
Nominal Pipe Size	Center to End, Elbows, Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread Min.	Width of Band, Min.	Inside Diameter of Fitting Min.	Inside Diameter of Fitting Max.	Metal Thickness, Min.	Outside Diameter of Band, Min.
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50	5.97	1.88	1.88	12.750	12.850	0.800	15.47
14 O.D.	10.40	....	2.00	2.00	14.000	14.100	0.880	16.94
16 O.D.	11.82	....	2.20	2.20	16.000	16.100	1.000	19.30

All dimensions given in inches.

grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follow:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure.

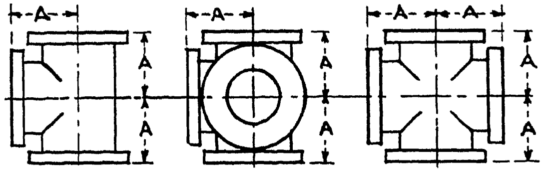
Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps,

TABLE 7. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON FLANGED FITTINGS



NOMINAL PIPE SIZE [a-b]	*A* CENTER TO FACE TEES AND CROSSES [b-c]	*AA* FACE TO FACE TEES AND CROSSES [b-c]	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN.	METAL THICKNESS OF BODY, MIN.
1	3½	7	4¼	7/16	7/16
1¼	3¾	7½	4⅝	½	7/16
1½	4	8	5	9/16	7/16
2	4½	9	6	⅝	7/16
2½	5	10	7	11/16	7/16
3	5½	11	7½	¾	7/16
3½	6	12	8½	13/16	7/16
4	6½	13	9	15/16	½
5	7½	15	10	15/16	½
6	8	16	11	1	9/16
8	9	18	13½	1⅛	⅝
10	11	22	16	1 3/16	¾
12	12	24	19	1¼	13/16
14 O.D.	14	28	21	1⅜	⅞
16 O.D.	15	30	23½	1 7/16	1
18 O.D.	16½	33	25	1 9/16	1 1/16
20 O.D.	18	36	27½	1 11/16	1⅛
24 O.D.	22	44	32	1⅞	1¼
30 O.D.	25	50	38¾	2⅛	1 7/16
36 O.D.	28	56	46	2⅜	1⅝
42 O.D.	31	62	53	2⅝	1 13/16
48 O.D.	34	68	59½	2¾	2

All dimensions given in inches.

[a]Size of all fittings listed indicates nominal inside diameter of port.

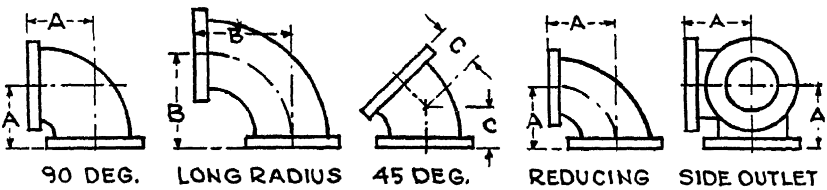
[b]Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[c]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand threads are used. To facilitate drainage, some elbows have the thread

TABLE 8. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR 125 LB CAST-IRON FLANGED FITTINGS



NOMINAL PIPE SIZE[a]	A: CENTER TO FACE ELBOW [b-c-d]	B: CENTER TO FACE LONG RADIUS ELBOW [b-c-d]	C: CENTER TO FACE 45 DEG ELBOW [c]	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN.	METAL THICKNESS OF BODY, MIN.
1	$3\frac{1}{2}$	5	$1\frac{3}{4}$	$4\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{16}$
$1\frac{1}{4}$	$3\frac{3}{4}$	$5\frac{1}{2}$	2	$4\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{16}$
$1\frac{1}{2}$	4	6	$2\frac{1}{4}$	5	$\frac{9}{16}$	$\frac{7}{16}$
2	$4\frac{1}{2}$	$6\frac{1}{2}$	$2\frac{1}{2}$	6	$\frac{5}{8}$	$\frac{7}{16}$
$2\frac{1}{2}$	5	7	3	7	$\frac{11}{16}$	$\frac{7}{16}$
3	$5\frac{1}{2}$	$7\frac{3}{4}$	3	$7\frac{1}{2}$	$\frac{3}{4}$	$\frac{7}{16}$
$3\frac{1}{2}$	6	$8\frac{1}{2}$	$3\frac{1}{2}$	$8\frac{1}{2}$	$\frac{13}{16}$	$\frac{7}{16}$
4	$6\frac{1}{2}$	9	4	9	$\frac{15}{16}$	$\frac{1}{2}$
5	$7\frac{1}{2}$	$10\frac{1}{4}$	$4\frac{1}{2}$	10	$\frac{15}{16}$	$\frac{1}{2}$
6	8	$11\frac{1}{2}$	5	11	1	$\frac{9}{16}$
8	9	14	$5\frac{1}{2}$	$13\frac{1}{2}$	$1\frac{1}{8}$	$\frac{5}{8}$
10	11	$16\frac{1}{2}$	$6\frac{1}{2}$	16	$1\frac{3}{16}$	$\frac{3}{4}$
12	12	19	$7\frac{1}{2}$	19	$1\frac{1}{4}$	$\frac{13}{16}$
14 O.D.	14	$21\frac{1}{2}$	$7\frac{1}{2}$	21	$1\frac{3}{8}$	$\frac{7}{8}$
16 O.D.	15	24	8	$23\frac{1}{2}$	$1\frac{7}{16}$	1
18 O.D.	$16\frac{1}{2}$	$26\frac{1}{2}$	$8\frac{1}{2}$	25	$1\frac{9}{16}$	$1\frac{1}{16}$
20 O.D.	18	29	$9\frac{1}{2}$	$27\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{1}{8}$
24 O.D.	22	34	11	32	$1\frac{7}{8}$	$1\frac{1}{4}$
30 O.D.	25	$41\frac{1}{2}$	15	$38\frac{3}{4}$	$2\frac{1}{8}$	$1\frac{7}{16}$
36 O.D.	28	49	18	46	$2\frac{3}{8}$	$1\frac{5}{8}$
42 O.D.	31	$56\frac{1}{2}$	21	53	$2\frac{5}{8}$	$1\frac{13}{16}$
48 O.D.	34	64	24	$59\frac{1}{2}$	$2\frac{3}{4}$	2

All dimensions given in inches.

[a]Size of all fittings listed indicates nominal inside diameter of port.

[b]Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[c]Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45 deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90 deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

[d]Side outlet elbows shall have all openings on intersection center-lines.

tapped at an angle to provide a pitch of the connecting pipe of ¼ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$-inch raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a $\frac{1}{16}$-inch raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 6, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 7 and 8.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a relatively narrow recessed facing.

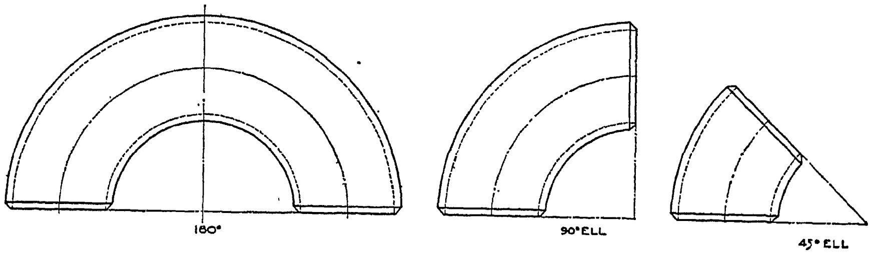
WELDING

Erection of piping in heating and ventilating installations by means of fusion welding has been commonly accepted in the past few years as a competitive method to the screwed and flanged joint. Since the question of economy of welding as against the use of screwed and flanged fittings is dependent on the individual job, the use of welding is generally recommended on the basis of a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding.

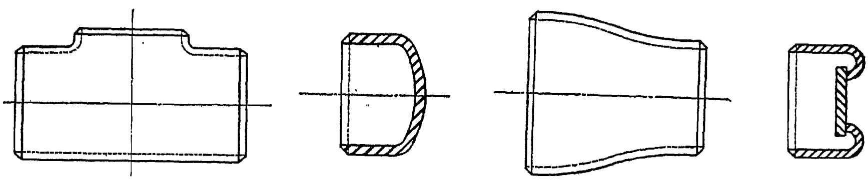
Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous know-

ledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests.



a. TYPICAL SHORT RADIUS ELBOWS



b. TEE | *c.* FORGED CAP | *d.* CONCENTRIC REDUCER | *e.* END CLOSURE

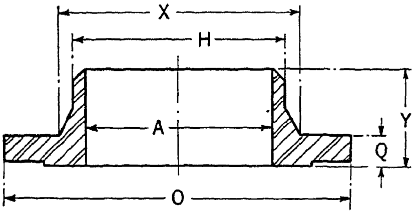
FIG. 1. TYPICAL WELDING FITTINGS

Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available with plain ends machine beveled for welding and with radii similar to short and long radius flanged fittings. Some typical types of these fittings

TABLE 9. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB AT 750 F



NOMINAL PIPE SIZE	DIAMETER OF FLANGE *O*	THICKNESS OF FLG. MIN. *Q*	DIAMETER OF HUB *X*	HUB DIAM. BEGINNING OF CHAMFER *H*	LENGTH THRU HUB *Y*	DIAM. FOR STANDARD PIPE *A*	DIAM. OF BOLT CIRCLE	NO. OF BOLTS	SIZE OF BOLTS
1	4¼	7/16	1 15/16	1.32	2 3/16	1.05	3⅛	4	½
1¼	4⅝	½	2 5/16	1.66	2¼	1.38	3½	4	½
1½	5	9/16	2 9/16	1.90	2 7/16	1.61	3⅞	4	½
2	6	⅝	3 1/16	2.38	2½	2.07	4¾	4	⅝
2½	7	11/16	3 9/16	2.88	2¾	2.47	5½	4	⅝
3	7½	¾	4¼	3.50	2¾	3.07	6	4	⅝
3½	8½	13/16	4 13/16	4.00	2 13/16	3.55	7	8	⅝
4	9	15/16	5 5/16	4.50	3	4.03	7½	8	⅝
5	10	15/16	6 7/16	5.56	3½	5.05	8½	8	¾
6	11	1	7 9/16	6.63	3½	6.07	9½	8	¾
8	13½	1⅛	9 11/16	8.63	4	7.98	11¾	8	¾
10	16	1 3/16	12	10.75	4	10.02	14¼	12	⅞
12	19	1¼	14⅜	12.75	4½	12.00	17	12	⅞
14 O. D.	21	1⅜	15¾	14.00	5	13.25	18¾	12	1
16 O. D.	23½	1 7/16	18	16.00	5	15.25	21¼	16	1
18 O. D.	25	1 9/16	19⅞	18.00	5½	17.25	22¾	16	1⅛
20 O. D.	27½	1 11/16	22	20.00	5 11/16	19.25	25	20	1⅛
24 O. D.	32	1⅞	26⅛	24.00	6	23.25	29½	20	1¼

All dimensions given in inches.

A raised face of 1/16 in. is included in *thickness of flange minimum*.

It is recommended that the taper of the hub should not exceed 6 degrees for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

are shown in Fig. 1. They are made in pipe sizes ¾ to 24 in., standard and extra heavy, in steel, wrought iron, brass, copper, and special alloys.

Socket welding fittings shown in Fig. 1 are commercially available. A proposed American Standard containing dimensions of steel welding-neck flanges for pressures up to 1500 lb per square inch has been developed in A.S.A. Sectional Committee B16. Tables 9 and 10 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per square inch gage pressure.

VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger

sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum. These valves may be secured with either a rising or a non-rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

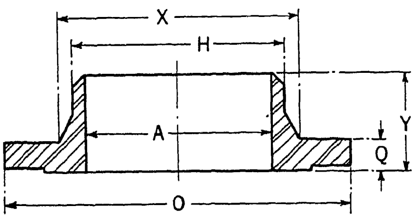
Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve in any position. Standard roughing-in dimensions for angle-type valves are given in Table 11.

Automatic control of steam supply to individual radiators can be effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a $\frac{1}{16}$-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly for use in hot water heating systems are of less complex design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the

TABLE 10. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 300 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 750 F



NOMINAL PIPE SIZE	DIAM. OF FLANGE	THICKNESS OF FLANGE MIN.	DIAM. OF HUB	HUB DIAM. BEGINNING OF CHAMFER	LENGTH THRU HUB	DIAM. FOR STANDARD PIPE	DIAM. FOR EXTRA STRONG PIPE	DIAM. OF BOLT CIRCLE	NO. OF BOLTS	SIZE OF BOLTS
	O	*Q*	*X*	*H*	*Y*	*A*	*A*			
*2	6½	⅞	3 5/16	2.38	2¾	2.07	1.94	5	8	⅝
2½	7½	1	3 15/16	2.88	3	2.47	2.32	5⅞	8	¾
3	8¼	1⅛	4⅝	3.50	3⅛	3.07	2.90	6⅝	8	¾
3½	9	1 3/16	5¼	4.00	3 3/16	3.55	3.36	7¼	8	¾
4	10	1¼	5¾	4.50	3⅜	4.03	3.83	7⅞	8	¾
5	11	1⅜	7	5.56	3⅞	5.05	4.81	9¼	8	¾
6	12½	1 7/16	8⅛	6.63	3⅞	6.07	5.76	10⅝	12	¾
8	15	1⅝	10¼	8.63	4⅜	7.98	7.63	13	12	⅞
10	17½	1⅞	12⅝	10.75	4⅝	10.02	9.75	15¼	16	1
12	20½	2	14¾	12.75	5⅛	12.00	11.75	17¾	16	1⅛
14 O. D.	23	2⅛	16¾	14.00	5⅝	13.25	....	20¼	20	1⅛
16 O. D.	25½	2¼	19	16.00	5¾	15.25	....	22½	20	1¼
18 O. D.	28	2⅜	21	18.00	6¼	17.25	....	24¾	24	1¼
20 O. D.	30½	2½	23⅛	20.00	6⅜	19.25	....	27	24	1¼
24 O. D.	36	2¾	27⅝	24.00	6⅝	23.25	....	32	24	1½

*For sizes below 2 inches use dimensions of 600 lb flanges.

All dimensions given in inches.

A raised face of 1/16 in. is included in *thickness of flange minimum.*

It is recommended that the taper of the hub should not exceed 6 degrees for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation ofsteam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system supplied with vacuum valves will heat more quickly and stay warm longer than one provided with straight pressure air valves; thus it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION[2]

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion in steam heating systems occurs principally in the condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaëration with commercial apparatus. The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaëration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible. The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton[3]. The exact application of this law, however, assumes equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

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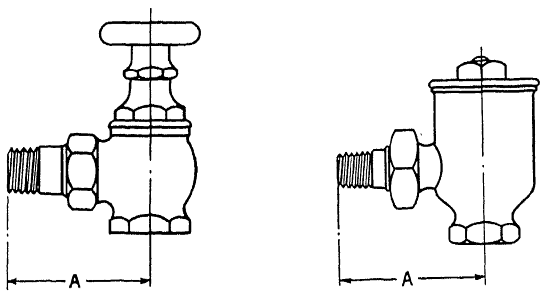
[2]New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933).

[3]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Distinction should be made between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam, such as water heaters, kitchen equipment, and sterilizers. Experience has shown that in heating systems the partial pressures of the gases do not reach such magnitudes as to cause harmful amounts of gas to become dissolved in the condensate when steam supplies are of reasonable purity. In other kinds of steam-using apparatus which are not ordinarily well vented, the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations. Consequently, corrosion is frequently observed in the condensate discharege lines from such apparatus, but this does not necessarily indicate that equally serious corrosion is taking place in the heating system supplied with steam from the same source.

TABLE 11. STANDARD ROUGHING-IN DIMENSIONS ANGLE TYPE VALVES



SIZE OF VALVE	DIMENSION A STEAM AND HOT WATER ANGLE VALVES AND UNION ELBOWS EFFECTIVE JANUARY 1, 1926	DIMENSION A MODULATING VALVES EFFECTIVE JANUARY 1, 1926	DIMENSION A RETURN LINE VACUUM VALVES EFFECTIVE JANUARY 1, 1925
1/2	2 1/4	2 3/4	3 1/4
3/4	2 3/4	2 3/4	......
1	3	3	......
1 1/4	3 1/2	3 1/2	......
1 1/2	3 3/4	3 3/4	......
2	4 1/4	4 1/4	......
Tolerance	±1/8	±1/8	......

All dimensions given in inches.

Connecting ends shall be threaded and gaged as to threading according to the American (Taper) Pipe Thread Standard, A.S.A. No. B2—1919.

The standardization of the Roughing-in Dimensions of Angle Steam and Hot Water, and Modulating Radiator Valves was made possible by the cooperation of the Manufacturers Standardization Society of the Valves and Fittings Industry.

When corrosive conditions are believed to exist, their seriousness should be determined by actual measurement, rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of

weight of the coils, referred to an established scale, indicates the relative corrosiveness of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaëration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Some success has been reported with the use of inhibitors, chief among which are oil, sodium silicate, and ammonia. Oil may be fed into the main steam-supply pipe by means of a sight-feed lubricator. The type of oil known as 600-W is usually recommended. In the present state of knowledge on this point, the quantity to be fed can best be determined by trial. The use of sodium silicate, fed in a similar manner, is reported to be successful but it has not been widely used.

The effect of ammonia is to increase the *pH* value of the condensate above the point where corrosion is likely to take place. Speller[4] reports having injected small quantities of ammonia into a small closed heating system (the entire amount of condensate being returned to the boiler) and finding the *pH* value maintained at a high point for several months without further additions of ammonia. The concentration of ammonia must be kept low to avoid corrosion of brass parts of the system. The use of ammonia is not to be recommended where steam may come in contact with food or other materials.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

[4]Corrosion in Steam Heating Systems, by F. N. Speller (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

PROBLEMS IN PRACTICE

1 ● What materials are used for pipes of heating systems?

Steel pipe is generally used for steam piping and for return lines where corrosion is not particularly active. Where corrosion is an important factor, it is good practice to use special corrosion-resistant ferrous alloys, or brass or copper pipe.

2 ● Why is thin-walled copper pipe made up with sweated joints?

If the pipe were threaded it would be necessary to use at least standard-weight wall thickness on account of the metal removed in threading. Flared ends with coupling nuts may be used, but this construction is expensive and hard to keep tight.

3 ● How are pipes designated in diameters of 12 in. and less?

By weight and nominal size, referring to the approximate inside diameter.

4 ● How are pipe sizes designated in diameters of 14 in. and more?

By wall thickness and outside diameter.

5 ● Why are expansion joints required in steam pipes?

To care for the change in length of the line brought about by a change in temperature.

6 ● What devices are used for taking up expansion?

Expansion joints, swivel joints, and the inherent flexibility of the pipe itself.

7 ● Where are swivel joints principally used?

In branch connections to radiators, and in the risers of multi-story buildings where they are installed between the floor joists.

8 ● Name three grades of American Standard screwed pipe fittings.

125-lb cast-iron, 150-lb malleable iron, and 250-lb cast-iron.

9 ● In what sizes are American standard cast-iron flanges and flanged fittings for 25-lb saturated steam pressure made?

In nominal sizes from 4 in. to 72 in., inclusive.

Chapter 35

WATER SUPPLY PIPING

Maximum Possible Flow, Maximum Probable Flow, Average Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Hot Water Storage

DOMESTIC water supply systems present the engineer with a design problem that requires combining the somewhat empirical rules and formulae in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same type of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulae employed may have been derived on sound technical bases the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such *zoning* or *sectionalizing* is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft[1].

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[1]It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions.

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

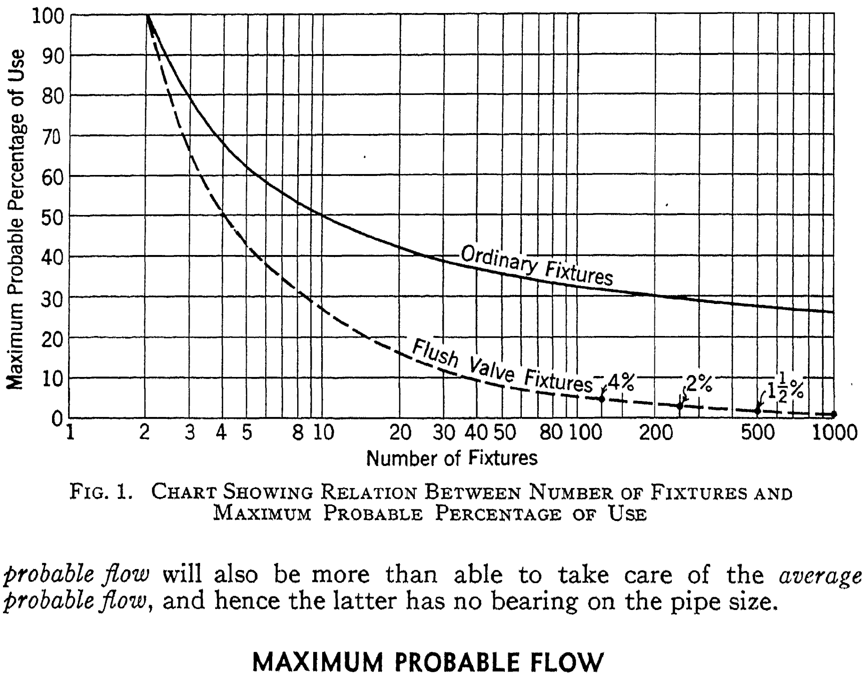


Fig. 1. Chart Showing Relation Between Number of Fixtures and Maximum Probable Percentage of Use

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time.* Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

Factor of Usage

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal. On the other hand it is probable that outside of flush valve

TABLE 1. APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

FIXTURES	COLD WATER (GALLONS PER MINUTE)	HOT WATER (GALLONS PER MINUTE)
Water-closets, flush valve	50[a]	0
Water-closets, flush tank	18	0
Urinals, flush valve	40[a]	0
Urinals, flush tank	18	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 5 to 6½ in. heads	3	3
Showers, tubular	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	4	4
Wash trays	3	3

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 50 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety. Urinal flush valves would vary proportionately in the same manner.

fixtures, the factor of usage would never be less than about 25 per cent no matter how many fixtures were installed, provided no fixtures in excess of those required for the actual occupancy were included.

This factor, beginning at 100 per cent for two ordinary fixtures, decreases rapidly until 5 fixtures are reached and then becomes almost constant, as shown in the upper curve, Fig. 1. This applies to a normal building and not to institutions where the inmates may all be required, for instance, to bathe on certain days of the week and at certain hours of those days. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved. For flush valve fixtures the quantity of water is greater, but owing to the short duration of the flush, the simultaneous usage drops more rapidly so as to reach 1 per cent for 1000 fixtures as shown, on lower curve, Fig. 1[2].

[2]This can be proved by assuming, for example, 1000 water-closets which would not be used more than six times per hour (or once every 10 minutes) and which require from 5 to 7 gal per flush or an average of about 6 gal. If these closets were all being used at their utmost capacity, the water demand would be 600 gpm. But average use would be about one-third of this and peak conditions would be in the neighborhood of twice the average, or about 400 gpm as the maximum that would ever develop. Assuming 50 gpm as the maximum rate of flow per closet and 1 per cent of the total closets in operation, the rate would be 50 gpm × 1 per cent of 1000 or 500 gpm. This is 100 gpm higher than obtained by the first method indicating an additional factor of safety over the first method.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water. Fig. 1 shows a maximum probable use for 50 water closets of about 8 per cent and for 150 ordinary fixtures, of about 31 per cent. Therefore:

Cold Water

50 W. C. x 50 gpm at 8 per cent		200 gpm
50 Lavs. x 3 gpm	150 gpm	
50 Sinks x 4 gpm	200 gpm	
50 Baths x 5 gpm	250 gpm	
150 Fixtures	600 gpm at 31 per cent	186 gpm
Total maximum probable flow of cold water		386 gpm

Hot Water

50 W. C.		None
50 Lavs. x 3 gpm	150 gpm	
50 Sinks x 4 gpm	200 gpm	
50 Baths x 5 gpm	250 gpm	
150 Fixtures	600 gpm at 31 per cent	186 gpm
Total for main supplying cold and hot water		572 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand.*

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used, and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantitities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than ¾ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43×100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30-lb drop under maximum probable demand is ample to take care of the friction caused by

the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

TABLE 2. SCHEDULE OF SIZES FOR DOWN-FEED RISER (SEE FIG. 2)

Portion of Riser	Maximum Probable Flow, Gallons per Minute																
	5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	¾	1	1¼	1¼	1½	1½	2	2	2½	2½	2½	2½	2½	3	3	3½	3½
S	¾	¾	1	1	1	1¼	1¼	1½	1½	1½	2	2	2	2	2½	2½	2½
R	¾	¾	¾	1	1	1	1¼	1½	1½	1½	2	2	2	2	2	2½	2½
Q	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
P	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
O	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
N	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
M	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
L	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
K	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
J	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
I	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
H	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
G	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
F	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
E	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
D	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
C	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
B	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
A	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½

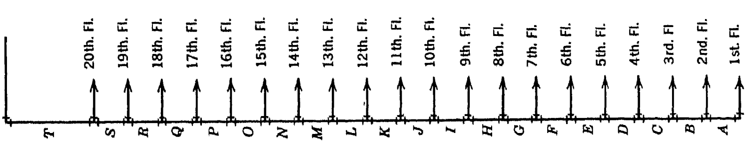


FIG. 2. TYPICAL RISER FOR 20-STORY BUILDING

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs; that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the

pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be

$$60 \text{ lb} - (15 \text{ lb} + 50 \text{ ft} \times 0.43 \text{ lb}) =$$
$$60 \text{ lb} - (15 \text{ lb} + 21.5 \text{ lb}) = 23.5 \text{ lb}$$

To change this into drop per 100 ft:

$$\frac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8 \text{ lb per } 100 \text{ ft.}$$

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

TABLE 3. APPROXIMATE ALLOWANCES FOR FITTINGS AND VALVES IN FEET OF STRAIGHT PIPE

Size of Pipe (Inches)	Type of Fitting or Valve					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
½	4	3	8	2	48	8
¾	5	3	10	3	60	10
1	5	3	10	3	60	10
1¼	6	4	12	3	72	12
1½	7	5	14	4	84	14
2	7	5	14	4	84	14
2½	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), or flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1 \text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be

$$0.43 \text{ lb} \times 20 \text{ ft} = 8.6 \text{ lb}$$

and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be

$$8.6 \text{ lb} - 1 \text{ lb} = 7.6 \text{ lb}$$

while the drop per 100 ft of equivalent run will have to be

$$\frac{1 \text{ lb} \times 100}{600} = 0.1667 \text{ lb.}$$

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8 in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe

would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

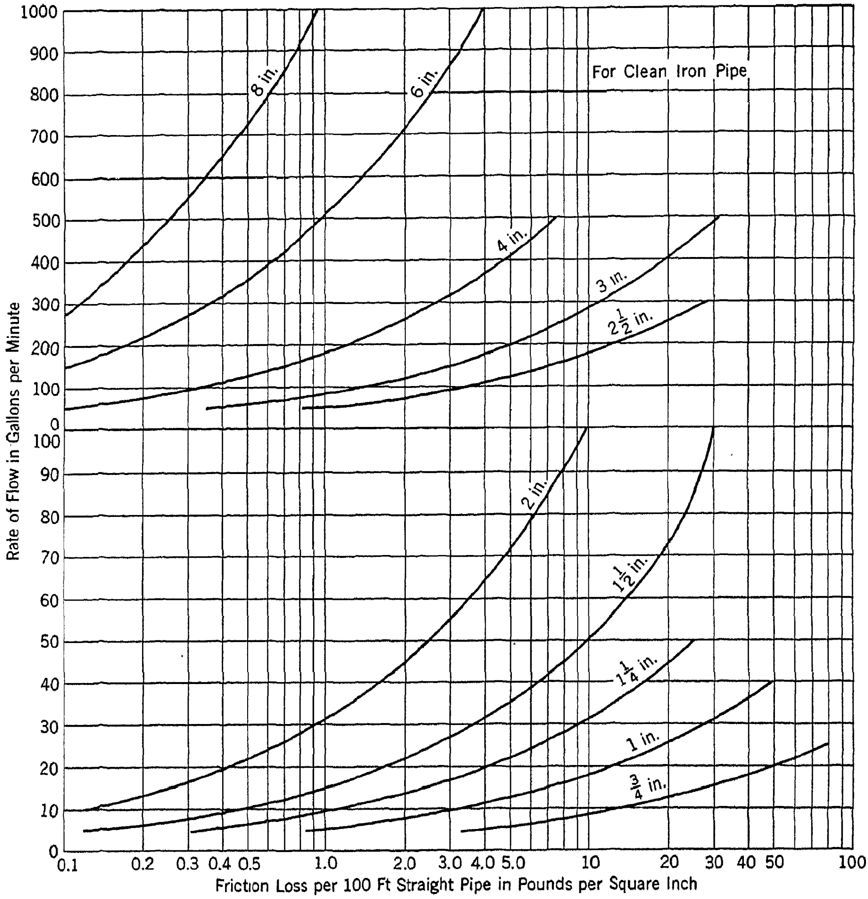


FIG. 3. CHART GIVING FRICTION LOSSES FOR VARIOUS RATES OF FLOW OF WATER

still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of

$$1 \text{ lb} + 4.3 \text{ lb} = 5.3 \text{ lb}$$

which, divided by the 600 ft equivalent run gives a drop per 100 ft of

$$\frac{5.3 \times 100}{600} = 0.9 \text{ lb}$$

and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 4 in., and 4 in.,

respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be

$$\frac{5.3 \text{ lb} \times 100}{200} = 2.7 \text{ lb per 100 ft.}$$

This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

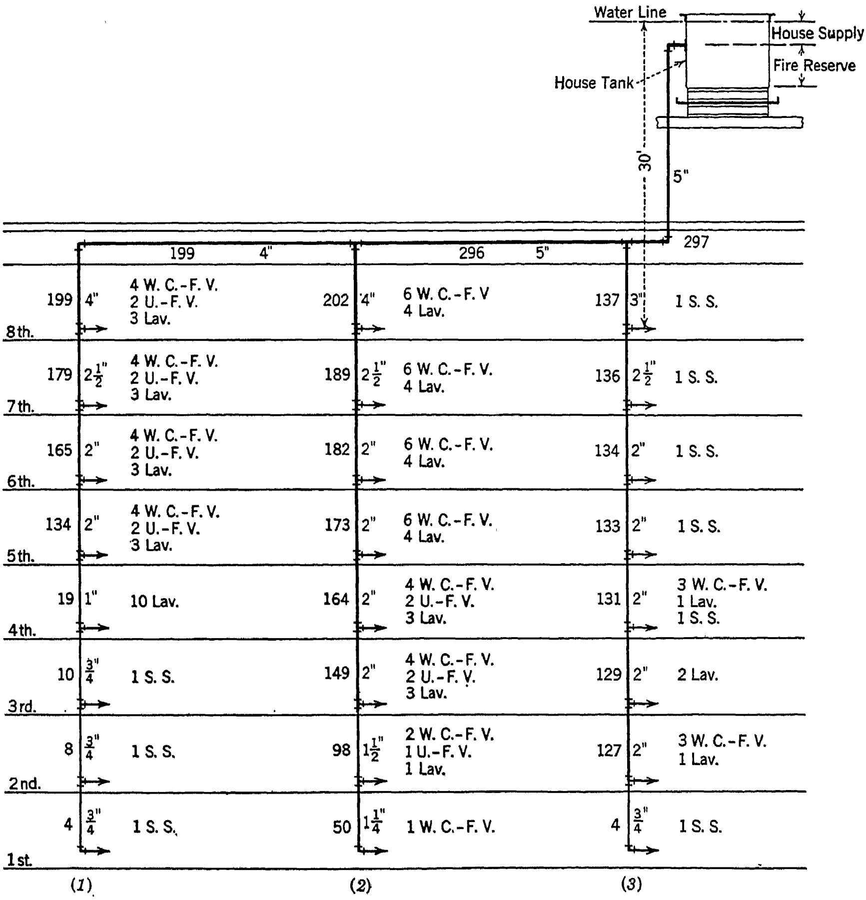


FIG. 4. TYPICAL LAYOUT FOR DOWN-FEED SYSTEM

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on

the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the

TABLE 4. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

Riser No. 1. (See Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM FIXTURES GPM	PROBABLE USE (PER CENT)	PROBABLE FIXTURES GPM	PROBABLE RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	100	4	4	30	¾
2nd	1 S. S.	4	4	100	4			
						8	30	¾
3rd	1 S. S.	4	4					
	3		12	80	10	10	30	¾
4th	10 Lav.	3	30					
	13		42	45	19	19	30	1
5th	4 W. C.	50	200					
	2 U.	40	80					
	6		280	40	112			
	3 Lav.	3	9					
	16		51	45	22	134	30	2
6th	4 W. C.	50	200					
	2 U.	40	80					
	12		560	25	140			
	3 Lav.	3	9					
	19		60	42	25	165	30	2
7th	4 W. C.	50	200					
	2 U.	40	80					
	18		840	18	151			
	3 Lav.	3	9					
	22		69	40	28	179	30	2½
8th	4 W. C.	50	200					
	2 U.	40	80					
	24		1120	15	168			
	3 Lav.	3	9					
	25		78	40	31	199	2	4

tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 4, 5, 6 and 7 show the schedule for Risers No. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 8 and 9.

This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem

TABLE 5. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

Riser No. 2. (See Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM FIXTURES GPM	PROBABLE USE (PER CENT)	PROBABLE FIXTURES GPM	PROBABLE RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 W. C.	50	50	100	50	50	30	1¼
2nd	2 W. C.	50	100					
	1 U.	40	40					
	4		190	50	95			
	1 Lav.	3	3	100	3	98	30	1½
3rd	4 W. C.	50	200					
	2 U.	40	80					
	10		470	30	141			
	3 Lav.	3	9					
	4		12	70	8	149	30	2
4th	4 W. C.	50	200					
	2 U.	40	80					
	16		750	20	150			
	3 Lav.	3	9					
	7		21	70	14	164	30	2
5th	6 W. C.	50	300					
	22		1050	15	157			
	4 Lav.	3	12					
	11		33	48	16	173	30	2
6th	6 W. C.	50	300					
	28		1350	12	162			
	4 Lav.	3	12					
	15		45	45	20	182	30	2
7th	6 W. C.	50	300					
	34		1650	10	165			
	4 Lav.	3	12					
	19		57	42	24	189	30	2½
8th	6 W. C.	50	300					
	40		1950	9	175			
	4 Lav.	3	12					
	23		69	40	27	202	2	4

would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 10 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up

$$0.43 \text{ lb} \times 80 = 34.4 \text{ lb}$$

and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the

Table 6. Typical Calculation of Pipe Sizes on Down-Feed Riser with Flush Valve Water-Closets and Urinals

Riser No. 3. (See Fig. 4)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Fixtures Gpm	Probable Use (per cent)	Probable Fixtures Gpm	Probable Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
1st	1 S. S.	4	4	100	4	4	30	¾
2nd	3 W. C.	50	150	80	120			
	1 Lav.	3	3					
	2		7	100	7	127	30	2
3rd	0 W. C.	0	000					
	3		150	80	120			
	2 Lav.	3	6					
	4		13	70	9	129	30	2
4th	3 W. C.	50	150					
	6		300	40	120			
	1 Lav.	3	3					
	1 S. S.	4	4					
	6		20	55	11	131	30	2
5th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	7		24	53	13	133	30	2
6th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	8		28	51	14	134	30	2
7th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	9		32	50	16	136	30	2½
8th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	10		36	48	17	137	2	3

street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as an even 300 ft. Then the allowable drop per 100 ft will be

$$\frac{25.6 \text{ lb} \times 100}{300} = 8.5 \text{ lb}$$

and the sizes shown in Fig. 5 are based on this amount of drop. Of course the other

risers will have the same maximum flows at the bottom as they formerly had at the top, namely 202 and 137 gal, respectively, for Risers No. 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 296 gpm, and between Riser No. 3 and the street main, 297 gpm which at a drop of 8.5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser

TABLE 7. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

Riser No.	Fixtures	GPM per Fixture	Maximum Fixtures GPM	Probable Use (Per Cent)	Probable GPM	Allowable Drop (Lb per 100 Ft)	Size of Main (Inches)
	16 W. C.	50	800				
	8 U.	40	320				
	24		1120	15	168		
1	22 Lav.	3	66				
	3 S. S.	4	12				
	25		78	40	31		
					199	2	4
	35 W. C.	50	1750				
	5 U.	40	200				
	64		3070	8	245		
2	23 Lav.	3	69				
	48		147	35	51		
					296	2	5
	6 W. C.	50	300				
	70		3370	7	236		
	4 Lav.	3	12				
3	6 S. S.	4	24				
	58		183	33	61		
					297	2	5

it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also because many plumbing fixtures have no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follow:

1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.

2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed.

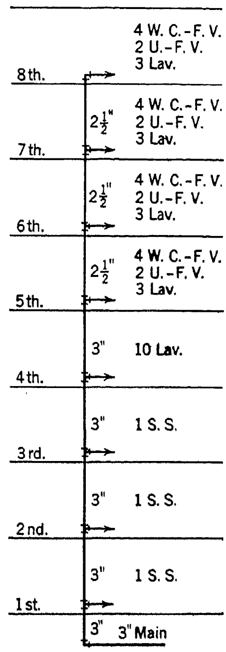
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

Table 8. Typical Calculation of Pipe Sizes on Down-Feed Risers with Flush Tank Water-Closets and Urinals on Top Floor Only (See Fig. 4)

Floor of Bldg.	Fixtures on Floor	gpm per Fixt.	Max. Fixt. gpm	Probable Use (per cent)	Probable Use gpm	Probable Riser gpm	Allowable Drop Lb per 100 ft	Pipe Size In.
				Riser No. 1				
7th and below	12 W. C.	50	600					
	6 U.	40	240					
	18		840	18	151			
	19 Lav.	3	57					
	3 S. S.	4	12					
	22		69	40	28	179	30	2½
8th	0 W. C.	00	000					
	0 U.	00	000					
	18		840	18	151			
	4 W. C.	18	72					
	2 U.	18	36					
	3 Lav.	3	9					
	31		186	37	69	220	3.3	4
				Riser No. 2				
7th and below	29 W. C.	50	1450					
	5 U.	40	200					
	34		1650	10	165			
	19 Lav.	3	57	42	24	189	30	2½
8th	0 W. C.	00	000					
	0 U.	00	000					
	34		1650	10	165			
	6 W. C.	18	108					
	4 Lav.	3	12					
	29		177	38	67	232	3.3	4
				Riser No. 3				
7th and below	6 W. C.	50	300	40	120			
	5 S. S.	4	20					
	4 Lav.	3	12					
	9		32	50	16	136	30	2½
8th	0 W. C.	00	000					
	6		300	40	120			
	1 S. S.	4	4					
	10		36	48	17	137	3.3	3

The last method is usually the most satisfactory. (See Fig. 6.)

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve, set on a 45-deg angle, and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 7.



(1)

FIG. 5. UP-FEED SYSTEM

For the second arrangement of hot water risers, circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across onto the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold

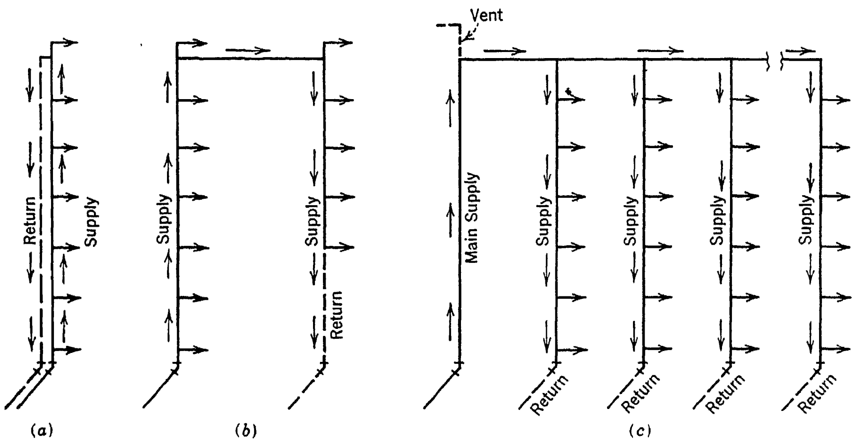


FIG. 6. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the botton of each riser should be arranged in the manner already outlined and any riser not going to the basement to

TABLE 9. SUMMARY OF RISER SIZES TO GIVEN MAIN SIZES (SEE FIG. 4)

RISER No.	FIXTURES	GPM PER FIXTURE	RISER GPM	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP (LB PER 100 FT)	SIZE OF MAIN (INCHES)
	12 W. C.	50	600				
	6 U.	40	240				
1	18		840	18	151		
	31 Fixt.		186	37	69		
					220	3.3	4
	29 W. C.	50	1450				
	5 U.	40	200				
2	52		2490	8	199		
	29 Fixt.		177				
	60		363	33	120		
					319	3.3'	4
	6 W. C.	50	300				
	58		2790	7	195		
3	10 Fixt.		36				
	70		399	33	131		
					326	3.3	4

TABLE 10. TYPICAL CALCULATION OF PIPE SIZES ON UP-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS (SEE FIG. 5)

Floor of Bldg.	Fixtures on Floor	Gpm per Fixture	Maximum Fixtures Gpm	Probable Use (per cent)	Probable Fixtures Gpm	Probable Riser Gpm	Allowable Drop Lb per 100 Ft	Pipe Size In.
			Riser No. 1					
8th	4 W. C.	50	200					
	2 U.	40	80					
	6		280	40	112			
	3 Lav.	3	9	80	7	119	8.5	2½
7th	4 W. C.	50	200					
	2 U.	40	80					
	12		560	25	140			
	3 Lav.	3	9					
	6		18	55	10	150	8.5	2½
6th	4 W. C.	50	200					
	2 U.	40	80					
	18		840	18	151			
	3 Lav.	3	9					
	9		27	50	14	165	8.5	2½
5th	4 W. C.	50	200					
	2 U.	40	80					
	24		1120	15	168			
	3 Lav.	3	9					
	12		36	47	17	185	8.5	3
4th	24 W. C.[a]							
	and U.		1120	15	168			
	10 Lav.	3	30					
	22		66	40	27	195	8.5	3
3rd	24 W. C.[a]							
	and U.		1120	15	168			
	1 S. S.	4	4					
	23		70	40	28	196	8.5	3
2nd	24 W. C.[a]							
	and U.		1120	15	168			
	1 S. S.	4	4					
	24		74	40	30	198	8.5	3
1st	24 W. C.[a]							
	and U.		1120	15	168			
	1 S. S.	4	4					
	25		78	40	31	199	8.5	3

[a]From floors above.

supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish

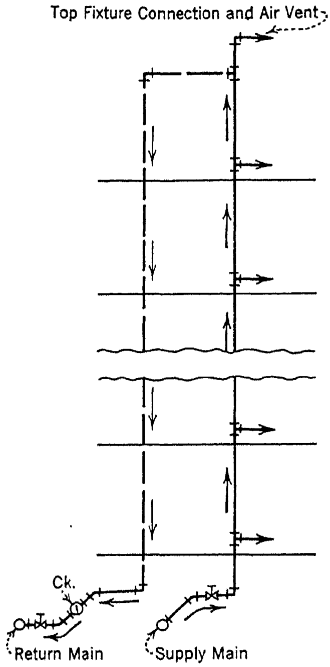


FIG. 7. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when

laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day is assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average.

HOT WATER STORAGE

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about

$$500 \times 40 \text{ gal} = 20{,}000 \text{ gal}$$

and this is an average of

$$\frac{20{,}000 \text{ gal}}{24} = 833 \text{ gph}$$

but the peak hour will require

$$\tfrac{1}{10} \text{ of } 20{,}000 = 2{,}000 \text{ gal}$$

and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be

$$2{,}000 - 833 = 1167 \text{ gal}$$

so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be

$$\frac{1167}{0.75} = 1556 \text{ gal}$$

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to say 1200 gph which then gives

$$2{,}000 - 1200 = 800 \text{ gal}$$

as the shortage during the peak hour, and the necessary storage will be

$$\frac{800 \text{ gal}}{0.75} = 1067 \text{ gal;}$$

or the heating capacity can be increased to 1500 gal, leaving a shortage of

$$2000 - 1500 = 500 \text{ gal}$$

TABLE 11. ORDINARY MAXIMUM HOURLY DEMAND FOR HOT WATER FOR VARIOUS FIXTURES IN GALLONS AND PROBABLE PERCENTAGE OF USAGE

TYPE OF BUILDING	LAVATORIES Private	LAVATORIES Public	BATHS	SHOWERS	SLOP SINKS	KITCHEN SINKS	PANTRY SINKS	FOOT BATHS	WASH TRAYS	AV. MAX USES[a]
MAXIMUM PROBABLE USAGE GPM	20	20	40	300	30	30	20	20	50	
	Probable Usage in Per Cent of Maximum Ordinary Use									
Apt. house	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gym.	25	100	100	100	------	----	------	100	------	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	------	100	------	90
Laundries	25	100	------	------	33	----	------	------	100	100
Office building	25	75	------	------	50	----	------	------	------	20
Baths	25	150	150	100	50	----	------	------	------	100
Residences	25	------	50	33	50	33	50	50	60	50
Schools	25	75	------	100	67	33	100	50	------	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

[a]Percentage of fixtures likely to be demanding maximum probable usage at any one time.

TABLE 12. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	
	(Transient)	40 (per day)
	(Men)	40 (per day)
	(Mixed)	60 (per day)
	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture).
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurants	$0.50 Meals	0.5 (per customer with hand washing)
		1.0 (per customer with machine washing)
	$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	$1.50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

and the storage required only

$$\frac{500}{0.75} = 667 \text{ gal.}$$

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage, as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 3½ sq ft[3] of equivalent steam heating surface (radiation) must be provided for every gallon of water heated 100 F or from 50 F to 150 F, which is the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (gph)	Additional Boiler Capacity (Sq Ft EDR)
833	2916
1200	4200
1500	5250

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 11 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 12 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

20 Baths at 40 gal and 33 per cent	270 gal
20 Lavs. at 20 gal and 25 per cent	100 gal
20 Sinks at 30 gal and 33 per cent	200 gal
20 Trays at 50 gal and 60 per cent	600 gal
Total	1170 gal
Probable peak use at one time	35 per cent
Probable actual peak demand	409 gph

[3]Actual requirement for 100-deg temperature difference $= \frac{100 \times 8.33}{240} = 3.33$ sq ft per gallon of water heated.

If three persons are assumed to an apartment the total daily use of hot water should approximate

$$20 \times 3 \times 40 \text{ gal} = 2400 \text{ gal}$$

and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal, which indicates that the values in Table 7 are safe.

PROBLEMS IN PRACTICE

1 ● Why is it impractical to size water supply piping so pipe friction will produce an equal pressure on each fixture?

Because the friction would be built up only in periods of maximum flow and at all other times it would be only a fraction of that required.

2 ● What is the purpose of zoning water supply systems in tall buildings?

To avoid excessive pressures in the lower stories.

3 ● Define the maximum possible flow, the maximum probable flow, and the average probable flow.

The maximum possible flow is the flow which would occur if all of the outlets on the system were opened at one and the same time. The maximum probable flow is the flow which will occur with probable peak conditions. The average probable flow is the flow likely to occur under a normal condition of use.

4 ● What is the factor of usage?

This is the percentage of the maximum possible flow which is likely to occur at peak load.

5 ● Within what limits does the factor of usage lie?

From 100 per cent for a single fixture down to about 28 per cent for 1000 fixtures of ordinary type, and from 100 per cent for a single fixture down to about 1 per cent for 1000 fixtures of the flush valve type.

6 ● How many feet higher than the uppermost fixtures must the water line in a house tank be to provide about 15 lb per square inch pressure at the fixture outlet?

Allowing for pipe losses, about 45 ft.

7 ● What methods of hot water circulation commonly are employed with hot water supply systems?

a. Upfeed risers with returns having no connections paralleling the risers.

b. Upfeed risers with returns in other locations, and with connections taken off both supply and return.

c. One main upfeed riser, without connections, supplying all downfeed risers for all fixtures.

8 ● Which method of hot water supply generally is the most satisfactory?

The single main upfeed riser supplying drop risers for all fixtures.

9 ● How much of the water stored in a hot water storage tank really is available for use?

About 75 per cent, because when only 25 per cent of the original water remains in the tank it has been so cooled down by the entering water that it is too cold for satisfactory use.

10 ● In cases of intermittent demand, does a large hot water storage tank increase or decrease the steam load for water heating?

It decreases the steam load in cases of intermittent demand but causes no change in the steam load if the demand is constant.

Chapter 36

INSULATION OF PIPING

Heat Losses from Bare Pipes, Steam and Hot Water Lines, Low Temperature Pipe Insulation, Pipe Sweating, Heat Losses from Pipe Surfaces, Thickness of Pipe Insulation, Underground Insulation

PIPE insulation performs an important function in preventing loss of heat where steam or hot water are conveyed from one part of a building to another, and in reducing the absorption of heat by cold pipes as well as preventing condensation on the outer surfaces.

BARE PIPE LOSSES

Heat losses from horizontal bare iron pipes, based on data obtained from tests conducted at the Mellon Institute, are given in Table 1. These losses are expressed in Btu per hour per linear foot of pipe per degree Fahrenheit difference in temperature between the steam or hot water in the pipe and the air surrounding the pipe. The monetary value of the loss of heat given in Table 1 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 1 and locate this value on the upper left hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right hand scale. In using this chart, the cost of coal should also include the labor for handling it, boiler room expense, etc. For additional information on this subject refer to paper entitled Heat Emission from Iron and Copper Pipe[1], by F. C. Houghten and Carl Gutberlet.

In order to determine heat losses per linear foot of pipe from known losses per square foot, it is necessary to know the area in square feet per linear foot of pipe. Table 2 gives these areas for various standard pipe sizes while Table 3 gives the area in square feet for flanges and fittings for various standard pipe sizes.

Very often, even where pipes are thoroughly insulated, flanges and fittings are left bare due to the belief that the losses from these parts are not large. However, the fact that a pair of 9-in. standard flanges having an area of 3.00 sq ft would lose, at 100 lb steam pressure, an amount of

[1]A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE IRON PIPES

Expressed in Btu per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.543	0.573	0.605	0.638	0.656	0.742	0.796
¾	0.660	0.690	0.729	0.762	0.781	0.886	0.955
1	0.791	0.829	0.878	0.920	0.953	1.084	1.166
1¼	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1½	1.09	1.15	1.220	1.29	1.335	1.520	1.640
2	1.34	1.40	1.491	1.58	1.637	1.866	2.015
2½	1.58	1.67	1.778	1.87	1.937	2.215	2.388
3	1.88	1.99	2.100	2.22	2.301	2.641	2.853
3½	2.13	2.24	2.380	2.51	2.585	2.972	3.215
4	2.36	2.50	2.650	2.78	2.873	3.312	3.582
4½	2.60	2.75	2.920	3.08	3.170	3.655	3.956
5	2.87	3.02	3.200	3.38	3.493	4.030	4.368
6	3.39	3.56	3.775	4.01	4.115	4.755	5.153
8	4.32	4.55	5.050	5.14	5.270	6.120	6.635
10	5.32	5.61	5.925	6.34	6.551	7.592	8.245
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

TABLE 2. RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

TABLE 3. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
10	3.43	5.20	10.18	13.58	12.35	15.60	15.41	20.41	19.58	26.26
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting.

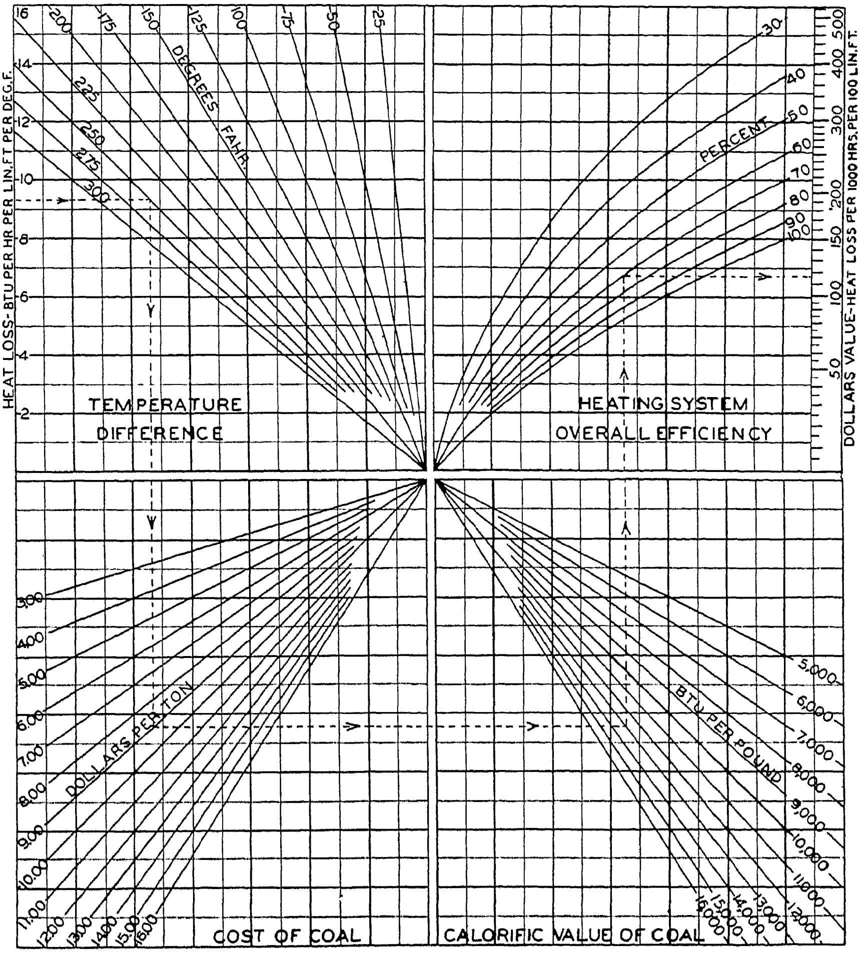


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 1)[a]

[a]This chart is based on 100 linear feet per 1000 hours. For fractions or multiples of these factors, multiply by proper percentage.

heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces. Table 3 shows the areas of both standard and extra heavy flanged fittings including the accompanying flanges bolted to the fittings.

STEAM AND HOT WATER LINES

The conductivities of various materials used for insulating steam and hot water pipes are given in Table 4. In this table the conductivities are given as functions of the mean temperatures or the mean of the inner and

TABLE 4. CONDUCTIVITIES (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

	MEAN TEMPERATURE				
	100 F	200 F	300 F	400 F	500 F
85 per cent Magnesia Type	0.425	0.465	0.505	0.550	0.590
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0.530	0.650	0.770	0.890	
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0.480	0.555	0.630	0.705	
Laminated Asbestos Type (30–40 Laminations per 1 in. thick)	0.360	0.415	0.470	0.525	0.585
Laminated Asbestos Type (20 Laminations per 1 in. thick)	0.545	0.605	0.665	0.725	0.785
Rock Wool Type	0.350	0.410	0.470	0.530	0.590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.515	0.545	0.575	0.605	0.635
Brown Asbestos Type (Felted Fibre)	0.600	0.640	0.675	0.715	0.750

[a]*Mechanical Engineers' Handbook*, Marks, 3rd Ed., 1930.

outer surface temperatures of the insulations. This method of stating conductivities makes it possible readily to calculate the heat loss through single or compound sections. It should be emphasized that the conductivities given in Table 4 for the various insulations are the average of values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through six of the types of insulation given in Table 4 for 1, 1½ and 2-in.-thick materials, and for temperatures commonly encountered in engineering practice can be obtained from Tables 5 to 10, inclusive. The loss through other thicknesses of the materials, and for other hot water or steam temperature conditions may be obtained by interpolation. The heat loss coefficients given in Tables 5 to 10 are based on the conductivities in Table 4 and were computed from data given in Chapter 22, THE GUIDE 1931.

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at low temperatures should be insulated so as to retard the flow of heat from the outside into the low temperature area and to prevent the formation of condensation and of frost if the temperatures are low enough, as well as to prevent corrosion induced by the presence of condensed moisture on metal surfaces. Materials commonly used for insulating pipes and surfaces at low temperatures are cork, rock cork, hair felt and other felted or fibrous non-absorbent materials. Thermal conductivities of low temperature insulating materials are given in Chapter 5.

Insulating materials are available commercially to meet varying temperature gradients. For example, the thickness of insulation for ice water is approximately 1½ in. if the temperature in the line is not lower than

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH 85 PER CENT MAGNESIA TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.744	0.754	0.764	0.774	0.779	0.802	0.814
	¾	0.672	0.681	0.689	0.697	0.701	0.721	0.731
	1	0.613	0.621	0.629	0.637	0.641	0.659	0.670
	1¼	0.562	0.570	0.577	0.585	0.589	0.606	0.617
	1½	0.532	0.539	0.546	0.553	0.557	0.573	0.582
	2	0.500	0.506	0.512	0.519	0.523	0.538	0.547
	2½	0.475	0.481	0.487	0.493	0.497	0.512	0.520
	3	0.455	0.461	0.467	0.474	0.477	0.492	0.500
	3½	0.441	0.447	0.452	0.458	0.462	0.475	0.483
	4	0.429	0.435	0.441	0.446	0.449	0.463	0.471
	4½	0.420	0.425	0.431	0.437	0.440	0.453	0.460
	5	0.411	0.416	0.422	0.427	0.430	0.443	0.450
	6	0.402	0.408	0.413	0.419	0.422	0.435	0.442
	8	0.387	0.392	0.397	0.403	0.405	0.418	0.425
	10	0.375	0.380	0.385	0.390	0.393	0.405	0.412
	12	0.369	0.374	0.378	0.383	0.386	0.398	0.405
1½	½	0.617	0.625	0.633	0.642	0.646	0.665	0.676
	¾	0.550	0.558	0.566	0.573	0.577	0.596	0.606
	1	0.496	0.503	0.511	0.518	0.522	0.540	0.549
	1¼	0.453	0.459	0.465	0.472	0.475	0.490	0.498
	1½	0.424	0.430	0.436	0.442	0.445	0.459	0.467
	2	0.394	0.400	0.405	0.410	0.413	0.427	0.434
	2½	0.371	0.376	0.382	0.386	0.389	0.401	0.408
	3	0.352	0.357	0.362	0.367	0.370	0.380	0.387
	3½	0.339	0.343	0.347	0.351	0.354	0.364	0.370
	4	0.328	0.333	0.337	0.341	0.343	0.353	0.359
	4½	0.320	0.324	0.328	0.332	0.334	0.343	0.350
	5	0.312	0.316	0.320	0.324	0.326	0.336	0.342
	6	0.303	0.307	0.311	0.315	0.318	0.328	0.333
	8	0.287	0.291	0.295	0.299	0.301	0.311	0.316
	10	0.276	0.280	0.284	0.288	0.290	0.299	0.304
	12	0.272	0.275	0.279	0.283	0.285	0.294	0.299
2	½	0.543	0.551	0.558	0.565	0.569	0.587	0.597
	¾	0.484	0.490	0.497	0.503	0.507	0.523	0.532
	1	0.433	0.439	0.445	0.451	0.454	0.467	0.476
	1¼	0.393	0.398	0.403	0.409	0.412	0.424	0.432
	1½	0.365	0.370	0.376	0.381	0.384	0.397	0.402
	2	0.338	0.343	0.347	0.351	0.354	0.364	0.370
	2½	0.316	0.320	0.324	0.328	0.331	0.341	0.347
	3	0.297	0.301	0.305	0.309	0.312	0.321	0.326
	3½	0.284	0.288	0.292	0.295	0.297	0.306	0.311
	4	0.275	0.278	0.282	0.285	0.287	0.296	0.301
	4½	0.266	0.270	0.273	0.276	0.278	0.286	0.290
	5	0.258	0.262	0.265	0.268	0.270	0.278	0.283
	6	0.250	0.254	0.257	0.260	0.262	0.270	0.274
	8	0.236	0.239	0.242	0.245	0.247	0.255	0.258
	10	0.224	0.227	0.230	0.233	0.235	0.242	0.246
	12	0.219	0.222	0.225	0.228	0.230	0.237	0.240

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (4 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.890	0.919	0.949	0.978	0.995	1.065	1.106
	¾	0.803	0.829	0.857	0.883	0.898	0.961	0.997
	1	0.731	0.756	0.780	0.804	0.818	0.876	0.909
	1¼	0.671	0.693	0.716	0.738	0.751	0.804	0.834
	1½	0.635	0.656	0.677	0.698	0.710	0.760	0.788
	2	0.595	0.615	0.635	0.656	0.667	0.715	0.742
	2½	0.567	0.586	0.605	0.624	0.635	0.680	0.705
	3	0.544	0.562	0.580	0.598	0.608	0.652	0.677
	3½	0.527	0.544	0.561	0.578	0.588	0.631	0.654
	4	0.513	0.530	0.548	0.565	0.575	0.616	0.639
	4½	0.502	0.518	0.535	0.551	0.561	0.601	0.624
	5	0.490	0.507	0.523	0.539	0.549	0.588	0.611
	6	0.480	0.496	0.512	0.528	0.538	0.577	0.599
	8	0.462	0.477	0.493	0.508	0.517	0.554	0.575
	10	0.447	0.462	0.476	0.491	0.500	0.537	0.557
	12	0.441	0.456	0.470	0.485	0.493	0.529	0.550
1½	½	0.737	0.762	0.787	0.812	0.826	0.884	0.918
	¾	0.657	0.679	0.702	0.725	0.737	0.790	0.820
	1	0.594	0.614	0.634	0.654	0.666	0.713	0.740
	1¼	0.542	0.559	0.577	0.596	0.606	0.649	0.673
	1½	0.507	0.524	0.541	0.558	0.568	0.609	0.632
	2	0.471	0.487	0.503	0.519	0.528	0.565	0.587
	2½	0.443	0.458	0.473	0.488	0.497	0.533	0.553
	3	0.421	0.435	0.449	0.463	0.472	0.506	0.525
	3½	0.403	0.417	0.430	0.443	0.451	0.483	0.502
	4	0.393	0.405	0.418	0.432	0.439	0.471	0.489
	4½	0.383	0.394	0.407	0.420	0.428	0.460	0.476
	5	0.372	0.384	0.397	0.409	0.417	0.447	0.463
	6	0.362	0.374	0.387	0.399	0.406	0.436	0.452
	8	0.343	0.354	0.366	0.378	0.385	0.413	0.429
	10	0.328	0.339	0.351	0.362	0.369	0.397	0.413
	12	0.323	0.334	0.346	0.357	0.364	0.391	0.407
2	½	0.648	0.670	0.692	0.713	0.726	0.779	0.810
	¾	0.578	0.598	0.617	0.637	0.648	0.694	0.720
	1	0.518	0.535	0.552	0.570	0.580	0.622	0.645
	1¼	0.469	0.485	0.501	0.517	0.527	0.566	0.587
	1½	0.438	0.452	0.467	0.481	0.490	0.526	0.545
	2	0.404	0.417	0.430	0.444	0.452	0.483	0.502
	2½	0.379	0.391	0.403	0.415	0.422	0.451	0.466
	3	0.356	0.367	0.378	0.390	0.397	0.425	0.440
	3½	0.339	0.350	0.361	0.373	0.380	0.406	0.421
	4	0.328	0.339	0.350	0.360	0.367	0.392	0.406
	4½	0.318	0.328	0.339	0.350	0.357	0.381	0.395
	5	0.308	0.318	0.329	0.340	0.346	0.370	0.384
	6	0.299	0.309	0.319	0.329	0.335	0.358	0.371
	8	0.282	0.291	0.301	0.310	0.315	0.336	0.349
	10	0.267	0.276	0.285	0.294	0.299	0.319	0.332
	12	0.263	0.272	0.280	0.289	0.294	0.314	0.325

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (8 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.801	0.820	0.838	0.857	0.868	0.913	0.939
	¾	0.723	0.739	0.756	0.773	0.783	0.824	0.847
	1	0.658	0.673	0.688	0.704	0.713	0.751	0.772
	1¼	0.606	0.619	0.633	0.647	0.655	0.688	0.707
	1½	0.573	0.586	0.599	0.612	0.619	0.652	0.670
	2	0.538	0.550	0.562	0.575	0.581	0.612	0.629
	2½	0.511	0.523	0.534	0.546	0.553	0.582	0.599
	3	0.489	0.501	0.512	0.524	0.531	0.558	0.575
1	3½	0.474	0.485	0.496	0.507	0.514	0.542	0.557
	4	0.461	0.472	0.482	0.493	0.500	0.527	0.542
	4½	0.451	0.462	0.472	0.482	0.489	0.515	0.530
	5	0.442	0.452	0.462	0.473	0.479	0.505	0.520
	6	0.432	0.442	0.452	0.463	0.468	0.493	0.508
	8	0.416	0.426	0.436	0.446	0.451	0.475	0.489
	10	0.402	0.412	0.421	0.430	0.435	0.459	0.473
	12	0.397	0.406	0.415	0.424	0.429	0.452	0.466
	½	0.664	0.679	0.695	0.711	0.720	0.759	0.780
	¾	0.593	0.607	0.621	0.636	0.643	0.677	0.697
	1	0.535	0.547	0.560	0.573	0.580	0.611	0.629
	1¼	0.488	0.499	0.510	0.522	0.528	0.556	0.572
	1½	0.457	0.467	0.478	0.490	0.496	0.522	0.537
	2	0.425	0.434	0.444	0.455	0.460	0.485	0.499
	2½	0.399	0.408	0.418	0.428	0.434	0.457	0.471
	3	0.378	0.387	0.396	0.405	0.411	0.433	0.446
1½	3½	0.363	0.371	0.380	0.388	0.393	0.415	0.427
	4	0.353	0.361	0.369	0.378	0.383	0.403	0.415
	4½	0.343	0.351	0.360	0.368	0.373	0.393	0.404
	5	0.334	0.342	0.350	0.358	0.363	0.383	0.394
	6	0.325	0.333	0.341	0.349	0.353	0.373	0.383
	8	0.309	0.316	0.324	0.332	0.336	0.355	0.365
	10	0.295	0.303	0.310	0.318	0.322	0.340	0.350
	12	0.291	0.298	0.306	0.313	0.317	0.335	0.344
	½	0.585	0.599	0.613	0.627	0.635	0.668	0.688
	¾	0.520	0.533	0.545	0.558	0.565	0.595	0.612
	1	0.465	0.476	0.487	0.498	0.504	0.532	0.547
	1¼	0.422	0.432	0.442	0.452	0.458	0.483	0.497
	1½	0.394	0.403	0.412	0.422	0.427	0.450	0.462
	2	0.364	0.372	0.380	0.388	0.393	0.415	0.427
	2½	0.339	0.347	0.355	0.363	0.367	0.387	0.398
	3	0.319	0.327	0.334	0.342	0.346	0.365	0.375
2	3½	0.304	0.311	0.318	0.326	0.330	0.349	0.358
	4	0.295	0.302	0.308	0.315	0.319	0.336	0.345
	4½	0.285	0.292	0.299	0.306	0.310	0.327	0.336
	5	0.278	0.284	0.290	0.297	0.301	0.317	0.326
	6	0.269	0.275	0.282	0.288	0.292	0.307	0.315
	8	0.253	0.259	0.265	0.270	0.273	0.288	0.296
	10	0.240	0.245	0.251	0.257	0.260	0.275	0.282
	12	0.236	0.241	0.247	0.253	0.256	0.270	0.277

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (30 TO 40 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water 120 F	Hot Water 150 F	Hot Water 180 F	Hot Water 210 F	Steam 227.1 F (5 Lb)	Steam 297.7 F (50 Lb)	Steam 337.9 F (100 Lb)
	Temperature Difference	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.605	0.620	0.635	0.650	0.658	0.695	0.716
	¾	0.546	0.560	0.573	0.586	0.594	0.627	0.645
	1	0.498	0.510	0.522	0.534	0.541	0.570	0.587
	1¼	0.457	0.468	0.480	0.491	0.497	0.525	0.540
	1½	0.432	0.442	0.453	0.464	0.470	0.496	0.511
	2	0.406	0.416	0.426	0.437	0.442	0.467	0.481
	2½	0.385	0.395	0.405	0.415	0.420	0.443	0.457
	3	0.370	0.379	0.389	0.398	0.403	0.425	0.438
	3½	0.359	0.367	0.376	0.385	0.390	0.413	0.426
	4	0.349	0.358	0.366	0.375	0.380	0.402	0.414
	4½	0.341	0.350	0.359	0.367	0.372	0.393	0.405
	5	0.334	0.342	0.351	0.359	0.364	0.384	0.395
	6	0.327	0.335	0.343	0.351	0.356	0.376	0.387
	8	0.314	0.322	0.330	0.338	0.343	0.362	0.373
	10	0.304	0.312	0.320	0.328	0.332	0.350	0.361
	12	0.301	0.308	0.316	0.324	0.328	0.346	0.356
1½	½	0.502	0.514	0.526	0.539	0.546	0.577	0.595
	¾	0.450	0.461	0.473	0.484	0.490	0.517	0.532
	1	0.405	0.415	0.426	0.436	0.442	0.466	0.480
	1¼	0.369	0.378	0.387	0.396	0.401	0.423	0.435
	1½	0.343	0.352	0.361	0.370	0.375	0.397	0.409
	2	0.321	0.329	0.337	0.345	0.350	0.369	0.380
	2½	0.301	0.309	0.317	0.324	0.330	0.348	0.358
	3	0.286	0.293	0.301	0.308	0.313	0.330	0.340
	3½	0.274	0.281	0.288	0.295	0.300	0.316	0.326
	4	0.267	0.273	0.280	0.287	0.291	0.307	0.317
	4½	0.259	0.266	0.272	0.279	0.283	0.299	0.308
	5	0.253	0.260	0.266	0.272	0.276	0.291	0.300
	6	0.247	0.253	0.260	0.266	0.269	0.284	0.293
	8	0.234	0.240	0.246	0.252	0.255	0.270	0.279
	10	0.223	0.229	0.235	0.241	0.245	0.258	0.266
	12	0.221	0.227	0.232	0.238	0.241	0.255	0.263
2	½	0.442	0.453	0.464	0.475	0.481	0.508	0.523
	¾	0.392	0.402	0.412	0.422	0.428	0.452	0.465
	1	0.352	0.360	0.369	0.378	0.383	0.405	0.417
	1¼	0.319	0.327	0.335	0.343	0.348	0.367	0.379
	1½	0.297	0.304	0.311	0.319	0.323	0.341	0.352
	2	0.274	0.280	0.287	0.294	0.298	0.314	0.324
	2½	0.256	0.262	0.269	0.275	0.279	0.293	0.302
	3	0.243	0.249	0.254	0.260	0.264	0.277	0.285
	3½	0.231	0.236	0.242	0.248	0.251	0.265	0.273
	4	0.223	0.228	0.234	0.240	0.243	0.257	0.265
	4½	0.216	0.222	0.227	0.233	0.236	0.249	0.256
	5	0.210	0.215	0.220	0.225	0.228	0.241	0.248
	6	0.203	0.208	0.213	0.218	0.221	0.233	0.240
	8	0.191	0.196	0.201	0.206	0.209	0.220	0.227
	10	0.182	0.187	0.192	0.196	0.199	0.210	0.215
	12	0.178	0.183	0.187	0.192	0.195	0.205	0.210

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (APPROXIMATELY 20 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.910	0.925	0.940	0.956	0.964	1.001	1.022
	¾	0.823	0.836	0.850	0.863	0.871	0.902	0.921
	1	0.748	0.760	0.773	0.785	0.792	0.823	0.840
	1¼	0.686	0.698	0.710	0.721	0.728	0.756	0.771
	1½	0.649	0.659	0.671	0.682	0.688	0.716	0.731
	2	0.610	0.620	0.630	0.640	0.647	0.671	0.685
	2½	0.581	0.590	0.600	0.609	0.615	0.638	0.651
	3	0.558	0.567	0.576	0.585	0.591	0.613	0.626
1	3½	0.539	0.548	0.557	0.566	0.571	0.592	0.604
	4	0.524	0.532	0.541	0.551	0.556	0.577	0.589
	4½	0.514	0.522	0.530	0.539	0.544	0.564	0.575
	5	0.503	0.511	0.519	0.528	0.533	0.553	0.565
	6	0.492	0.500	0.509	0.517	0.522	0.542	0.553
	8	0.473	0.480	0.488	0.497	0.502	0.521	0.532
	10	0.458	0.465	0.473	0.481	0.485	0.504	0.514
	12	0.452	0.459	0.467	0.475	0.478	0.497	0.507
	½	0.755	0.767	0.780	0.793	0.800	0.831	0.848
	¾	0.674	0.685	0.697	0.708	0.715	0.743	0.759
	1	0.607	0.618	0.628	0.639	0.645	0.670	0.684
	1¼	0.553	0.562	0.572	0.581	0.587	0.610	0.622
	1½	0.517	0.527	0.536	0.545	0.550	0.572	0.584
	2	0.481	0.490	0.499	0.508	0.513	0.535	0.547
	2½	0.453	0.460	0.469	0.477	0.481	0.500	0.511
	3	0.429	0.436	0.444	0.452	0.456	0.475	0.485
1½	3½	0.412	0.419	0.427	0.434	0.438	0.456	0.465
	4	0.400	0.407	0.415	0.422	0.426	0.443	0.453
	4½	0.390	0.396	0.402	0.409	0.413	0.429	0.437
	5	0.380	0.386	0.393	0.400	0.403	0.418	0.427
	6	0.369	0.375	0.382	0.389	0.392	0.408	0.417
	8	0.351	0.358	0.364	0.370	0.374	0.388	0.397
	10	0.337	0.344	0.350	0.356	0.359	0.373	0.382
	12	0.332	0.338	0.344	0.350	0.353	0.367	0.375
	½	0.664	0.675	0.687	0.698	0.704	0.732	0.747
	¾	0.591	0.601	0.611	0.621	0.627	0.652	0.665
	1	0.529	0.538	0.547	0.557	0.562	0.584	0.597
	1¼	0.480	0.488	0.497	0.505	0.510	0.529	0.540
	1½	0.445	0.453	0.462	0.470	0.475	0.494	0.504
	2	0.412	0.420	0.427	0.434	0.438	0.455	0.464
	2½	0.385	0.392	0.398	0.405	0.409	0.425	0.434
	3	0.364	0.370	0.376	0.382	0.385	0.400	0.408
2	3½	0.346	0.352	0.358	0.365	0.368	0.382	0.390
	4	0.336	0.342	0.348	0.354	0.357	0.371	0.378
	4½	0.325	0.332	0.338	0.343	0.346	0.360	0.367
	5	0.316	0.322	0.327	0.333	0.336	0.349	0.356
	6	0.306	0.312	0.317	0.323	0.326	0.338	0.345
	8	0.288	0.293	0.298	0.303	0.306	0.317	0.324
	10	0.275	0.279	0.284	0.289	0.292	0.302	0.308
	12	0.269	0.274	0.278	0.283	0.286	0.296	0.302

Table 10. Coefficients of Transmission (U) for Pipes Insulated with Rock Wool Type Insulation

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.631	0.644	0.658	0.672	0.680	0.712	0.730
	¾	0.569	0.581	0.593	0.606	0.613	0.642	0.659
	1	0.518	0.529	0.541	0.552	0.559	0.585	0.600
	1¼	0.476	0.486	0.497	0.507	0.513	0.537	0.551
	1½	0.450	0.460	0.470	0.480	0.485	0.508	0.522
	2	0.422	0.431	0.441	0.450	0.456	0.478	0.490
	2½	0.402	0.411	0.420	0.428	0.434	0.455	0.466
	3	0.385	0.394	0.402	0.411	0.415	0.435	0.446
	3½	0.373	0.381	0.389	0.398	0.402	0.421	0.432
	4	0.363	0.371	0.379	0.387	0.392	0.411	0.422
	4½	0.355	0.363	0.371	0.379	0.383	0.402	0.413
	5	0.348	0.356	0.364	0.371	0.376	0.394	0.404
	6	0.341	0.348	0.356	0.363	0.368	0.386	0.396
	8	0.327	0.335	0.342	0.349	0.353	0.372	0.381
	10	0.317	0.324	0.331	0.338	0.343	0.360	0.369
	12	0.313	0.320	0.327	0.334	0.338	0.355	0.364
1½	½	0.523	0.534	0.545	0.556	0.563	0.590	0.606
	¾	0.468	0.477	0.487	0.497	0.503	0.528	0.542
	1	0.421	0.430	0.440	0.449	0.455	0.477	0.490
	1¼	0.383	0.391	0.399	0.407	0.412	0.433	0.444
	1½	0.359	0.366	0.375	0.383	0.387	0.407	0.419
	2	0.333	0.340	0.348	0.356	0.360	0.378	0.389
	2½	0.314	0.320	0.327	0.335	0.339	0.355	0.365
	3	0.296	0.302	0.310	0.317	0.321	0.337	0.347
	3½	0.286	0.291	0.298	0.304	0.307	0.323	0.332
	4	0.278	0.284	0.290	0.296	0.300	0.315	0.323
	4½	0.270	0.276	0.282	0.287	0.291	0.305	0.313
	5	0.263	0.269	0.275	0.280	0.284	0.298	0.305
	6	0.257	0.262	0.267	0.273	0.277	0.290	0.297
	8	0.244	0.249	0.254	0.260	0.263	0.276	0.283
	10	0.235	0.240	0.245	0.250	0.253	0.265	0.272
	12	0.230	0.234	0.239	0.245	0.247	0.260	0.267
2	½	0.461	0.471	0.481	0.491	0.496	0.520	0.534
	¾	0.409	0.418	0.427	0.436	0.441	0.463	0.475
	1	0.366	0.374	0.382	0.390	0.395	0.415	0.427
	1¼	0.333	0.340	0.347	0.355	0.359	0.377	0.387
	1½	0.310	0.316	0.323	0.330	0.334	0.351	0.360
	2	0.286	0.292	0.298	0.304	0.308	0.323	0.331
	2½	0.268	0.274	0.279	0.285	0.289	0.302	0.310
	3	0.252	0.257	0.262	0.268	0.272	0.284	0.292
	3½	0.241	0.246	0.251	0.257	0.260	0.272	0.280
	4	0.232	0.237	0.242	0.247	0.250	0.262	0.269
	4½	0.225	0.230	0.235	0.240	0.243	0.255	0.262
	5	0.218	0.223	0.228	0.233	0.236	0.247	0.253
	6	0.213	0.217	0.221	0.226	0.228	0.239	0.245
	8	0.200	0.204	0.208	0.213	0.215	0.225	0.231
	10	0.189	0.193	0.197	0.201	0.204	0.214	0.220
	12	0.185	0.190	0.194	0.198	0.200	0.210	0.216

25 F; the thickness of insulation for brine is approximately 2½ in. where the temperature ranges from 0 deg to 25 F; and the thickness of insulation where the brine temperature ranges from −30 F to zero degrees is approximately 4 in.

Insulation To Prevent Freezing

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 11 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that the most efficient insulation be utilized. This table is based on the use of hair felt or cork, having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below the freezing point of water (temperature difference, 60 F).

The last column of Table 11 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column of Table 11. However, if the water enters the pipe at 34 F it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference, 80 F), instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 11, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all

TABLE 11. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES

Nominal Pipe Size (Inches)	Number of Hours to Cool Water to Freezing Point			Water Required to Flow to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches					
	1	2	3	1	2	3
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.44
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.46	1.93

been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water, but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 11, the only safe way to insure against freezing is to install a steam or hot water line, or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

Pipe Sweating

In some cases the prevention of condensation rather than the conservation of heat is the governing factor in determining the thickness of insulation required. Fig. 2 may be used for determining the thickness of any material of known conductivity which should be used to prevent condensation on pipes and flat metallic surfaces. The surface resistances used for calculating the family of curves in Fig. 2 are based on the results of tests made on canvas-covered pipe insulation surfaces at Mellon Institute. However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves given may be followed with no alteration for surfaces commonly used.

Moisture will be deposited on a surface whenever its temperature falls to that of the dew point. The maximum permissible temperature drop is indicated on Fig. 2 at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop

represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of condensation in Chapter 7.)

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces the increases in losses due to air velocity are very small as compared with increases shown for bare surfaces, because of the

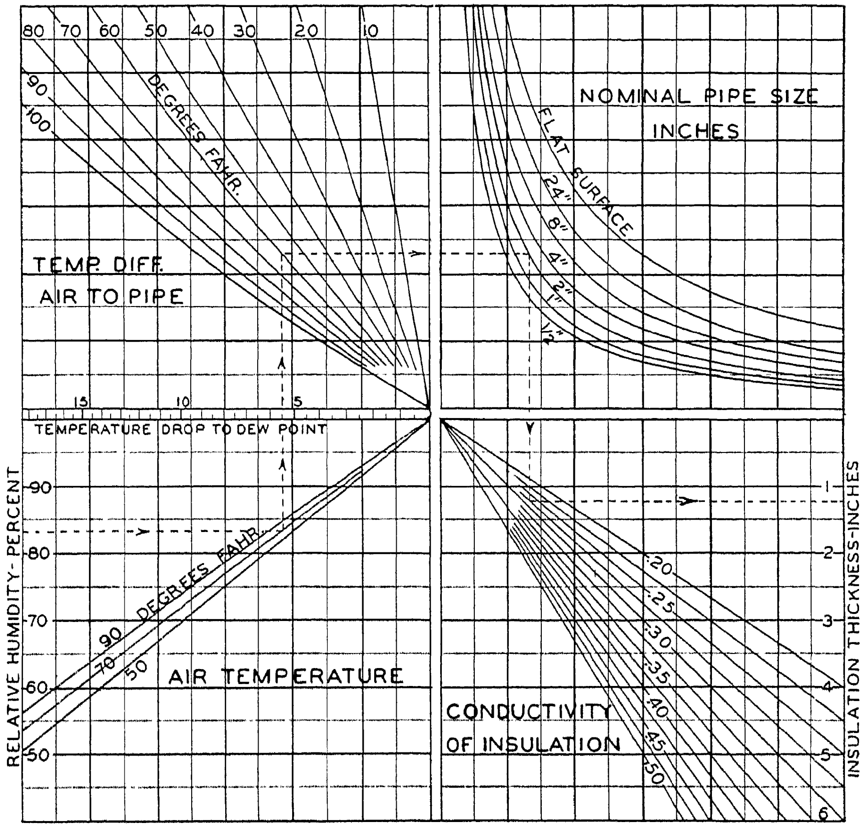
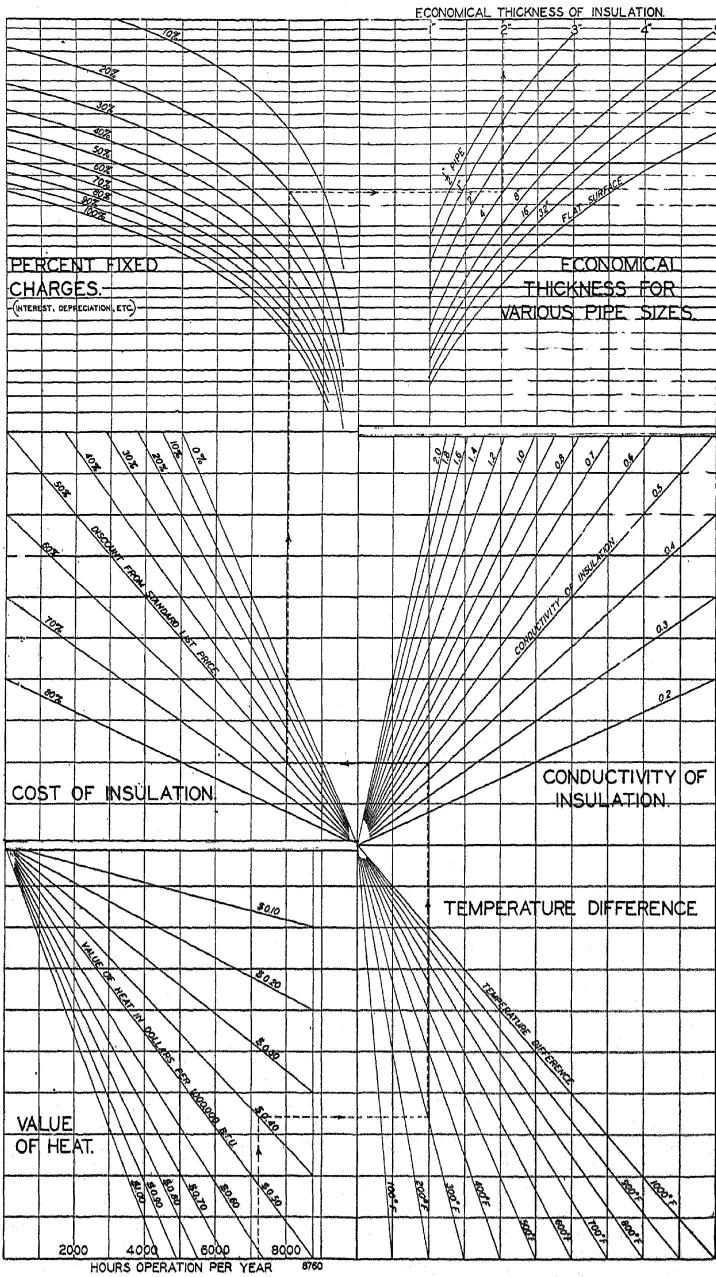


FIG. 2. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in loss due to air velocity ranges from about 30 per cent in the case of 1-in. thick insulation, to about 10 per cent in the case of 3-in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.



(L. B. McMillan, *Proc. National Dist. Heating Ass'n.*, Vol. 18, p. 131.)

FIG. 3. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as possible. Pipe insulation out-of-doors should be provided with a waterproof jacket, and other outdoor insulation should be thoroughly weatherproofed.

ECONOMICAL THICKNESS OF PIPE INSULATION

Table 12 shows the thicknesses of insulation which ordinarily are used for various temperature conditions. Where a thorough analysis of economic thickness is desired, this may be accomplished through the use of the chart, Fig. 3.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally, to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of

TABLE 12. THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally, to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly. The dotted line on the chart illustrates its use in solving a typical example.

Underground Insulation

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 37.) Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical

[2]*Handbook of the National District Heating Association*, Second Edition, 1932.

protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult of accurate determination due to the many variables which have to be considered. As a result of theories developed by J. R. Allen[3], together with experimental data presented by others, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 13 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 3. Fig. 3 is based on conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

TABLE 13. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Lb Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

[3]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920).

PROBLEMS IN PRACTICE

1 ● Compute the total annual heat loss from 165 ft of 2-in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained from Table 7, Chapter 1. The temperature difference between the pipe and air = 239.4 − 70 = 169.4 deg. By interpolation of Table 1 between temperature differences of 157.1 F and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4

deg is found to be 1.677 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire = 1.677 × 169.4 × 165 (linear feet) × 4000 (hours) = 188,000,000 Btu.

2 ● Coal costing $11.50 per ton and having a calorific value of 13,000 Btu per pound is being burned in the furnace supplying steam to the pipe line given in Question 1. If the system is operating at an over-all efficiency of 55 per cent determine the monetary value of the annual heat loss from the line.

The cost of heat per 1 million Btu supplied to the system = 1,000,000 × 11.5 (dollars) ÷ 13,000 (Btu) × 2000 (lb) × 0.55 (efficiency) = $0.804. The total cost of heat lost per year = 0.804 × 188 (million Btu) = $151.15.[4]

3 ● If the steam line given in Question 1 is covered with 1-in. thick 85 per cent magnesia, determine the resulting total annual heat loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

By interpolation of Table 5 between temperature differences of 157.1 F and 227.7 F, the coefficient of transmission for 1-in. magnesia on a 2-in. pipe is found to be 0.525 Btu per hour per square foot of pipe surface per degree temperature difference at a temperature difference of 169.4 deg. The total hourly loss per square foot of insulated pipe will then be 0.525 × 169.4 = 89.04 Btu. From Table 2 the area per linear foot of 2-in. pipe is found to be 0.622 sq ft. The total annual loss through the insulation = 89.04 × 0.622 × 165 (linear feet) × 4000 (hours) = 36,550,000 Btu. The annual bare pipe loss as determined in the solution of Question 1 was found to be 188,000,000 Btu. The saving due to insulation is then 188,000,000 − 36,550,000 = 151,350,000 Btu per year.

From the solution of Question 2 it was found that the heat supplied to the system cost $0.804 per million Btu; therefore, the monetary value of the saving = 0.804 (dollars) × 151.35 (million Btu) = $121.69, or 81.2 per cent of the cost when using uninsulated pipe.

4 ● The manufacturer's list price for 85 per cent magnesia insulation is $0.36 per linear foot for 1-in. (standard thick) material to cover a 2-in. pipe. Determine the period of time required for the saving found in Question 3 to pay for the cost of the insulation if it can be purchased and applied at 80 per cent of list price (20 per cent discount).

The applied cost of insulation = 165 (linear feet) × 0.36 (dollars) × 0.80 (net) = 47.52. Since the annual saving as found in Question 3 amounts to $121.69, the insulation will pay for its cost in 47.52 ÷ 121.69 = 0.3905 years; in other words, the cost will be repaid 2.56 times by the saving obtained in one heating season.

5 ● The conductivity of magnesia insulation is 0.455 at the mean temperature which will result under the conditions of Question 3. Estimate the most economical thickness of magnesia for application on the pipe when operating under the conditions which are given in the foregoing problems and when a 20 per cent return is required on the investment for insulation.

Use chart given in Fig. 3. Begin at the left bottom margin and proceed successively as shown by the dotted line example to the following essential data which are collected from the problems previously given:

4000 hours operation per year.
$0.804 value of heat, dollars per million Btu.
169.4 deg temperature difference.
0.455 conductivity of insulation.
20 per cent discount from list, cost of insulation.
20 per cent fixed charges, return on investment.
2-in. pipe size.

Solution of the problem by use of Fig. 3 results in a required thickness of approximately

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[4]A closely approximate solution of this problem may be quickly made by use of the estimating chart given in Fig. 1.

1.05 in. The nearest commercial thickness procurable is standard thick ($1\frac{1}{32}$ in.) magnesia.

(It is of interest to note that the use of Fig. 3 will generally result in solutions which, for all practical purposes, agree closely with the specifications for thicknesses given in Table 12.)

6 ● Determine the minimum thickness of wool felt insulation having a conductivity of 0.30 necessary to prevent condensation of moisture on a 4-in. pipe carrying cold water at a temperature of 40 F when the surrounding air reaches maximum conditions of 90 F with a relative humidity of 90 per cent.

The difference between the temperature of the pipe and the surrounding air is $90 - 40 = 50$ deg. For quick estimating purposes use the chart given in Fig. 2. Enter this chart at the lower left margin on the 90 per cent relative humidity line and proceed horizontally to the right to intersect the 90 deg air temperature line. Project a line up to the 50 deg temperature difference line, and then horizontally to the right to the intersection with the 4-in. pipe size line. From this point proceed down to intersect the 0.30 line which denotes the conductivity of the insulation. Directly opposite this point of intersection the correct thickness of insulation is read from the scale on the lower right margin. This chart solution denotes that wool felt 2.4-in. thick is sufficient to prevent condensation. The nearest commercial thickness procurable is $2\frac{1}{2}$ in.

For prevention of condensation as well as for protection against freezing, if the thickness determined theoretically cannot be had, it is better to apply the next greater thickness procurable rather than to use any lesser thickness because an additional factor of safety is thus obtained.

7 ● A 3-in. pipe covered with 2 in. of hair felt insulation carries water out-of-doors. Weather Bureau records for the locality denote that a minimum outdoor temperature of zero F may be expected to prevail for a period not to exceed 10 hours. By use of Table 11 determine what degree of protection is provided against freezing if the water is stationary in the line for the 10 hour period.

Table 11 denotes that 4.27 hours are required to lower water temperature from 42 F to the freezing point (a 10 F drop) when the initial temperature difference between the water and air is 60 F. The temperature drop in the example is $45 - 32 = 13$ F and the temperature difference is $45 - 0 = 45$ F. The time required to lower the water from 45 F to the freezing point will therefore be $4.27 \times \frac{13}{10} \times \frac{60}{45} = 7.4$ hours.

It is evident that insufficient protection is provided to prevent freezing if a temperature of zero F prevails outside for any period longer than 7 hours 24 min.

8 ● With data given in Question 7 and its solution, determine the minimum flow of water which must be maintained to prevent freezing if the length of the water line is 85 ft.

From Table 11 it is seen that a flow of 0.94 lb of water per linear foot per hour is sufficient to prevent a drop in the water temperature from 42 F to the freezing point when the temperature difference between air and water is 60 F. With the conditions stated, a flow of $0.94 \times \frac{10}{13} \times \frac{45}{60} = 0.542$ lb of water per linear foot of pipe line per hour, or $85 \times 0.542 = 46.07$ lb of water per hour must flow through the line in order to prevent freezing.

Chapter 37

DISTRICT HEATING

Underground Steam Piping, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Pipe Conduits, Pipe Tunnels, Service Connections, Steam per Square Foot of Heating Surface, Fluid Meters and Metering, Rates

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories and for the design of heating systems for buildings which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

UNDERGROUND STEAM PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Any unusual requirements such as those for process steam should be individually calculated.

The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating

peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend, in part, upon whether or not it has been passed through electrical generating units. If it has, the pressure will be considerably lower than if live steam, direct from the boilers, is used. The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, and (3) simpler problems in pressure reduction at the buildings. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators. The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\, W^2 L \left(1 + \frac{3.6}{d}\right)}{y d^5} \qquad (1)$$

where

P = pressure drop, pounds per square inch.
W = weight of steam flowing, pounds per minute.
L = length of pipe, feet.
d = inside diameter of pipe, inches.
y = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 32.

Information on provision for expansion will be found in Chapters 32 and 34.

In general, return lines when installed follow the contour of the land, and Table 1 gives sizes of return pipes for various grades. It is evident that at points where the grade is great, smaller pipes can be installed.

PIPE CONDUITS

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural steel set in concrete.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft.						
	6″	1′	2′	3′	5′	10′	20′
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	....
8	190000	277000	404000	508000	660000	938000	....
10	344000	498000	724000	900000	1190000	....	....
12	555000	798000	1148000	1499000	1990000	....	....

[a]Size of pipe should be increased if it carries any steam.

In laying out conduits of this type the following points should be borne in mind:

1. An expansion joint offset or bend should be placed between each two anchors.
2. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart.
3. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.
4. For longer lines, manholes must be located according to judgment and depending upon the expansion value of the type of expansion joint or bend that is used. The minimum number of manholes will be required when an expansion bend or an anchor with double expansion joint is placed in each manhole and the pipes are anchored midway between manholes.
5. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic pressure should be one-and-one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around

the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at *A*. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At *B* and *D* in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation.

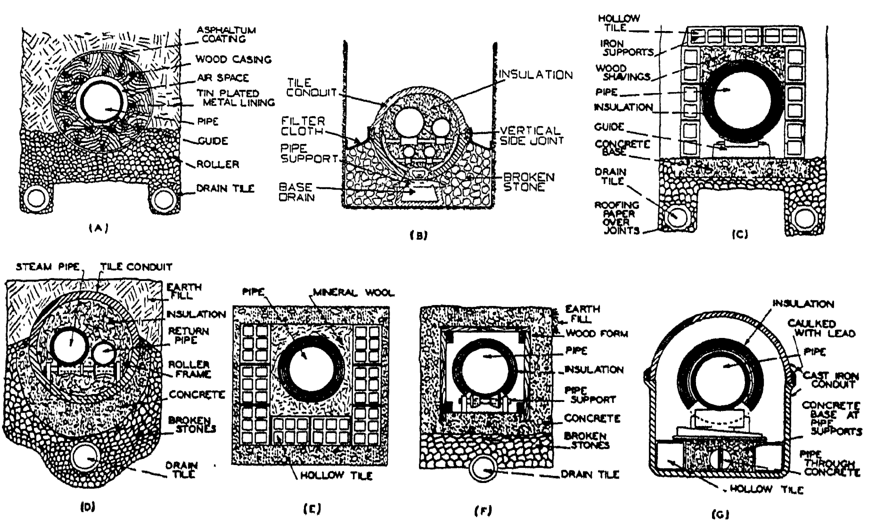


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in

separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduits.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a waterproof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required

to accommodate miscellaneous other services or provide underground passage between buildings.

SERVICE CONNECTIONS

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer.

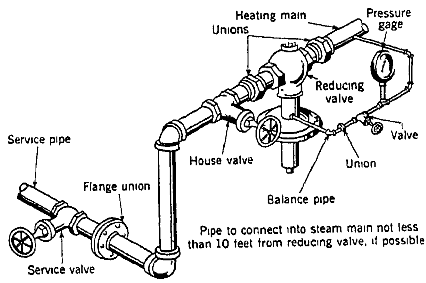


FIG. 2. CONNECTIONS FOR REDUCING VALVES OF SIZE LESS THAN 4 INCHES

There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

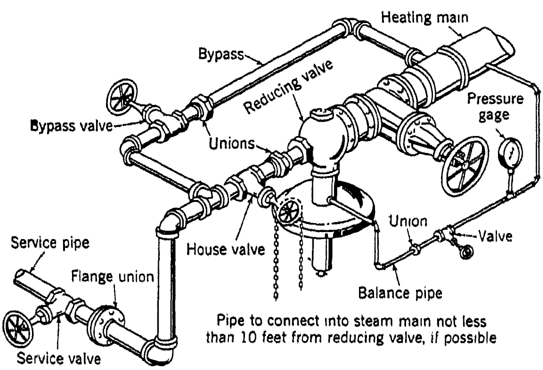


FIG. 3. CONNECTIONS FOR REDUCING VALVES OF SIZE 4 INCHES AND LARGER, AND FOR EXPANDED VALVES

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. This latter construction permits the

operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve, usually furnished by the utility company,

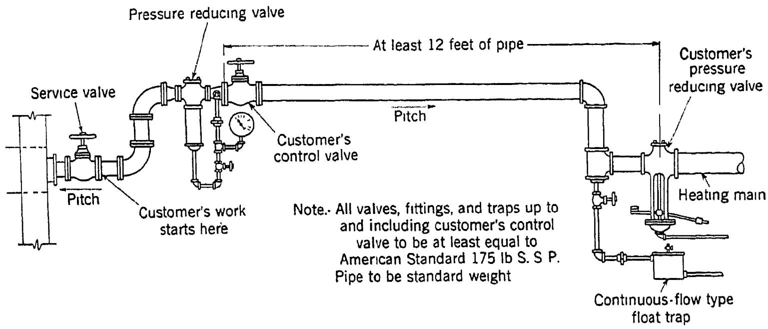


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING CONDENSATION METER

effects the initial pressure reduction. The second reducing valve, usually furnished by the customer, reduces the steam pressure to that required.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in

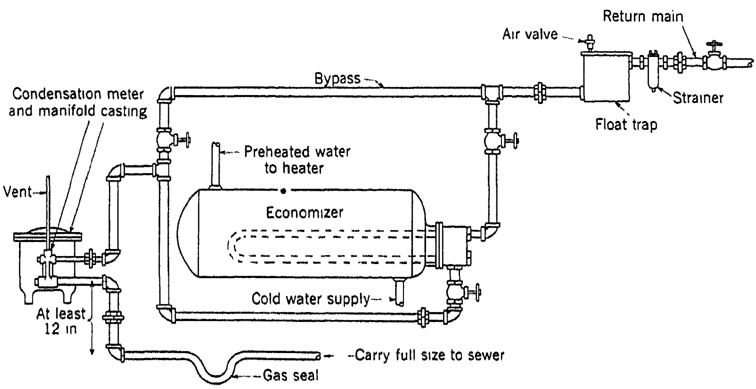


FIG. 5. RETURN PIPING FOR CONDENSATION METER

some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very

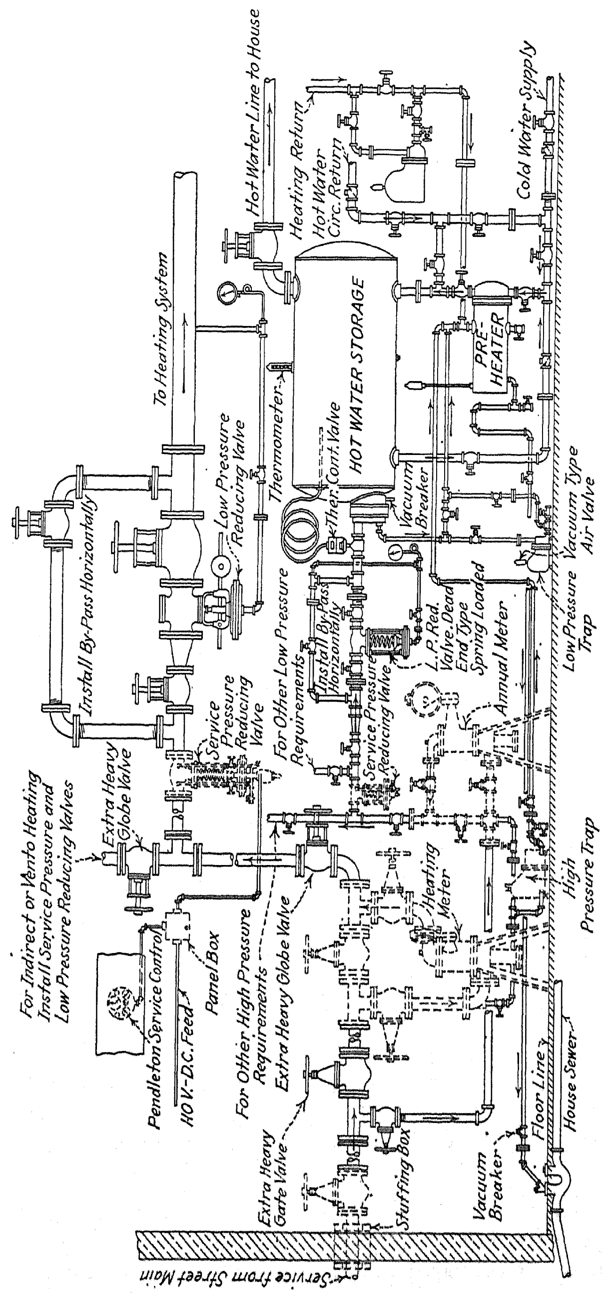


FIG. 6. TYPICAL SERVICE INSTALLATION

cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling radiator, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the preheater on its way to the meter. The supply to the hot water heater passes through the preheater, absorbing heat from the condensation. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the preheater and the water heater proper, not at the preheater inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the preheater should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensation if storage capacity is provided for the preheated water. Frequently a type of preheater is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of thermostats of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating thermostat. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiation either throughout the 24 hours of the day or during the day-

time hours only. The setting of a switch may provide no service, continuous service, or periodic service. For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature. A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the operation of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

FLUID METERS

No one thing has contributed more to the advancement of district heating than the perfection of fluid meters, which may be classified as follows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the stream and isolated by alternately filling and emptying containers of known capacity.

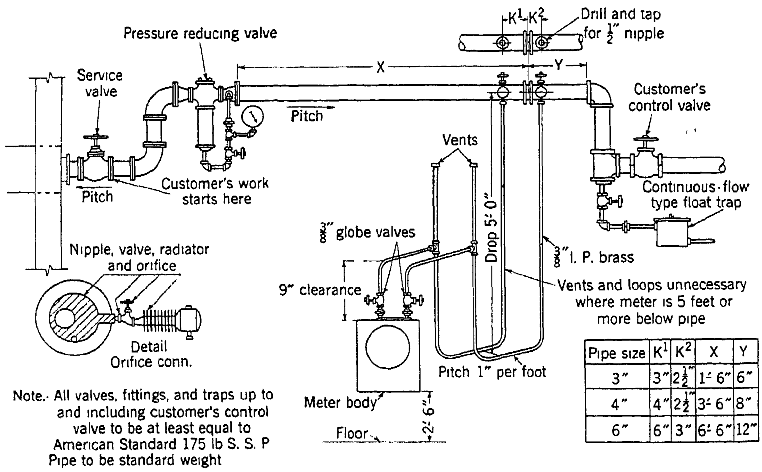
2. *Differential Meters:* The fluid does not pass in isolated separately-counted quantities but in a continuous stream which may flow through the line without actuating the primary device of the meter. In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional subdivisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.



Pipe size	K^1	K^2	X	Y
3"	3"	2½"	1'-6"	6"
4"	4"	2½"	3'-6"	8"
6"	6"	3"	6'-6"	12"

FIG. 7. ORIFICE METER STEAM SUPPLY CONNECTION

6. Cleanliness of the fluid to be measured.
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow: turbulent, pulsating, or steady.
10. Cost.
    (*a*) Purchase price.
    (*b*) Installation cost.
    (*c*) Calibration cost.
    (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer.
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing.
14. Stocking of repair parts.
15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are of the condensation or flow types.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Steam flow meters are available in many types and combinations, as indicated in the subdivision covering fluid meters on page 648.

The *orifice and plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a

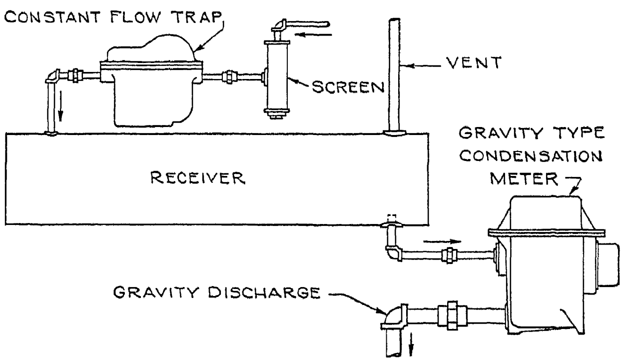


FIG. 8. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 7 shows a typical orifice type meter connection and indicates typical requirements in the installation of this type of meter. Fig. 8 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 9 shows a vacuum installation without a master trap.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.

3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.
4. Meter piping should be kept free from leaks.
5. Sludge should not be permitted to collect in the meter body.
6. The meter body and meter piping should be kept above freezing temperatures.
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM PER SQUARE FOOT OF HEATING SURFACE

The following factors are used in New York City for the different classes of buildings listed. The factors are based on maintaining an inside temperature of 70 F for certain hours, with a minimum outside temperature of 0 F and an average of 43 F for the heating season of eight months (October 1 to June 1). In this group are six types of buildings:

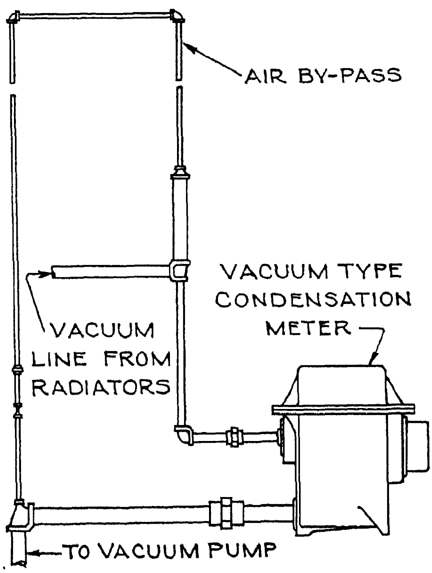


FIG. 9. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

Manufacturing or commercial loft type where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 a.m. to 5 p.m. No Sunday or holiday use and no night use. *Factor:* 325 lb per square foot of heating surface per season.

Office buildings using steam during daylight hours to maintain 70 F from 9 a.m. to 6 p.m. for approximately 240 days (heating season). No night use. *Factor:* 400 lb per square foot of heating surface per season.

Office buildings using steam during day hours and at night when required to 7, 8 and 9 p.m. (customary where there are stock brokers or banking offices), 240 days. *Factor:* 500 lb per square foot of heating surface per season.

Residences of the block type (not detached) where high-class heating service is required; somewhat similar to apartment buildings. *Factor:* 550 lb per square foot of heating surface per season.

Apartment houses where high-class heating service is required. (Steam off at midnight.) *Factor:* 650 lb per square foot of heating surface per season.

Hotels (commercial type) where very high-class service is required for 24 hours. *Factor:* 800 lb per square foot of heating surface per season.

By assuming one square foot of equivalent heating surface for each 100 cu ft of space heated, which seems a fair ratio in New York City, it is possible roughly to estimate the steam required per cubic foot of space, information which is often more easily obtained than the square feet of heating surface. Additional data on the heating requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. The profit need not be divided proportionately among the rated groups, but should be established from a competitive standpoint. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average load to the maximum load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per hour per square foot of surface.

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

A. Flat Rates.
  1. Radiator surface charge. *Obsolescent.*

B. Meter Rates.
  1. Straight-line.
  2. Step. *Obsolescent.*
  3. Block.
    (*a*) Class rates.

C. Demand Rates.
  1. Flat demand.
  2. Wright.
  3. Hopkinson.
  4. Doherty (or Three charge).

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate. The pounds of steam consumed by a customer are divided into blocks of M pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefited at the expense of the others.

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time.

> The *flat demand rate* is usually expressed in dollars per M lb of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a flow meter is not practicable.
>
> The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.
>
> The *Hopkinson demand rate* is divided into two elements:
>
> (*a*) A charge based upon the demand, either estimated or measured;
>
> (*b*) A charge based upon the amount of steam consumed.
>
> This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand. They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per M lb of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

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PROBLEMS IN PRACTICE

1 ● What is the common method of determining the size of mains in a distribution system?

On the basis of pressure drop: The initial pressure and the minimum permissible terminal pressure are specified, and the pipe sizes are so chosen that the maximum estimated amount of steam may be transmitted without exceeding this pressure difference. The steam's velocity is disregarded and it may reach a magnitude in excess of 35,000 fpm which is not considered high.

2 ● a. What are the advantages and disadvantages of a low pressure distribution system?

b. High pressure?

a. The advantages of a low pressure distribution system include:

1. Smaller heat loss from the pipes.
2. Less trouble with traps and valves.
3. Simpler problems with pressure reducing equipment at the buildings.
4. No danger to building heating equipment from high pressure through failure of the reducing valves.

The disadvantages of a low pressure system are:

1. Larger pipe sizes.
2. Decreased field of usefulness owing to small pressure range.

b. The advantages of a high pressure system are:

1. Smaller pipe sizes.
2. Greater adaptability of the steam to various uses other than building heating.

The disadvantages of a high pressure system are:

1. Large heat loss from the pipes.
2. The high pressure traps and valves required often give more trouble than low pressure traps and valves do.
3. Extra heavy fittings are required.
4. Usually two reducing valves or some form of emergency relief is necessary to protect the building piping system.

3 ● Determine the size of pipe from the following data using Unwin's formula:
Length of pipe, 600 ft.

Steam to be carried, 90,000 lb per hour, dry saturated.

Initial pressure, 100 lb per square inch, gage.

Final pressure, 40 lb per square inch, gage.

Using the formula:

$$P = \frac{0.0001306\ W^2 L \left(1 + \frac{3.6}{d}\right)}{y\ d^5}$$

The pressure drop $P = 100 - 40 = 60$ lb per square inch.

The weight of steam per minute $W = \frac{90,000}{60} = 1500.$

The length of pipe in feet $L = 600.$

The average density of steam y in pounds per cubic foot, taken from Keenan's Table:

At 100-lb gage, $y = 0.2578$
At 40-lb gage, $y = 0.1285$
Average, $y = 0.1932$

The diameter of the pipe in inches $= d.$

Substituting the values in the formula:

$$60 = \frac{0.0001306 \times 1500^2 \times 600 \left(1 + \frac{3.6}{d}\right)}{0.1932 \times d^5}$$

$$d = 7.35 \text{ in.}$$

Therefore, an 8-in. pipe should be used.

4 ● What points should be borne in mind when laying out an underground steam conduit?

The conduit should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or the conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit.

5 ● What is considered the proper pressure for a hydrostatic test before completing the conduit?

In the case of any underground piping which is to be buried or otherwise made inaccessible, the assembled lines shall first be tested hydrostatically at a pressure of one and one-half times the maximum allowable service pressure and held for a period of at least two hours without evidence of leakage. In any case the hydrostatic pressure should not be less than 100 lb per square inch.

6 ● What factors should be considered before determining the route of a steam line?

1. The line should be so located that it will bring in the greatest revenue (or supply the most steam) with the least cost.
2. The ultimate length and size of services and branches necessary with each possible location should be estimated, for mains should be run near to the big loads.
3. The location of the boiler room or piping center of present and future buildings to be served should be considered.
4. Where possible, make the lines straight between manholes.
5. Avoid such obstructions as other lines, sewers, ducts, curb drains, manholes, valve boxes, catch basins, fire hydrants, and poles; especially avoid electric ducts and water lines.
6. Avoid locating lines near where pile driving and foundation construction for new buildings will take place.
7. Consider construction difficulties such as traffic, hard rock, and wet earth, which increase time and labor.
8. Consider the economies of using available sidewalk vaults of buildings. Weigh the advantage of less excavation against the cost of obstruction removal.
9. Consider all operating difficulties.
10. Consider the difficulties of negotiating agreements for lines on private property where public and private rights-of-way are available.
11. Consider the effect of proposed municipal and other improvements.
12. Consider municipal regulations.

7. ● State the advantages and disadvantages of tunnels over conduits.

The advantages of pipe tunnels over conduits are:

1. Accommodation for miscellaneous services other than steam.
2. Provision of an underground passage between buildings.
3. Easy installation of additional pipes and easy replacement of existing pipes with larger sizes.
4. Easy inspection and maintenance of pipes.

The disadvantages of pipe tunnels over conduits are:

1. Higher first cost.
2. Higher maintenance cost in general.

8 ● Is the steam consumption less in a building that shuts off its steam at night than in one that does not? Why?

It has been thoroughly demonstrated that the steam consumption is less in a building where the steam is shut off at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

9 ● What are the common methods for salvaging heat in condensate?

The most common methods are:

1. The use of a water heating economizer for preheating the hot water supply to the building.
2. The use of a cooling radiator.

10 ● What are the common means used to graduate the heat supply according to variations in outside temperature?

a. A weather compensating thermostat regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply; at the same time it insures delivery of steam to all the radiators.

b. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds to produce some control over the heat output.

c. The use of an orifice system graduates heat supply.

d. The time-limit control which may be set to provide no service, continuous service, or periodic service, is also used. For periodic service, steam may be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. This type of service is provided by several intermittent settings. A night switch will maintain the intermittent day setting, or interrupt the day operation and cut off the supply of steam at night during any desired hours.

Chapter 38

RADIANT HEATING

Physical and Physiological Considerations, British Equivalent Temperature, Control of Heat Losses, Methods of Application, Principles of Calculation, Mean Radiant Temperature, Measurement of Radiant Heating

HEATING for comfort is generally understood to mean that heat must be supplied to control the rate of heat loss from the human body so that the physiological reactions are conducive to a feeling of comfort in the individual. While in convection heating, as described in Chapter 30, heat is transferred from a heating unit to the air and thence to the occupant, the primary object of radiant heating is to warm the occupant directly without heating the air to any extent. Thus, the difference between convection heating and radiant heating is partly physical and partly physiological.

Comfort requires that heat be removed from the body at the same rate as it is generated by the oxidation of the foodstuffs in the body tissues. The normal rate of heat production in a sedentary individual is about 400 Btu per hour[1], or, since the entire surface area of an average adult is 19.5 sq ft, about 20.5 Btu per square foot per hour. Conditions should be such as to remove heat at this rate if the surface is to be maintained at the mean normal surface temperature of the human body.

Heat is transferred from any warm dry body to cooler surroundings principally by convection and by radiation, the approximate total rate of heat loss being the sum of the two. Where the body surface is moist there is additional loss of heat through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the difference between the temperature of the body and that of the surrounding air, and on the rate of air motion over the body. The loss by radiation depends entirely upon the difference between the temperature of the body and the mean surface temperature of the surrounding walls and objects. This latter temperature is called the *mean radiant temperature* (MRT). Because these two types of heat loss act in a supplementary manner toward each other, a required rate of heat loss can be secured by having a relatively low air temperature and a relatively high MRT, or *vice versa*. Thus, if the air is reduced from a given temperature to a lower temperature, the amount of heat lost from the body by convection is increased, and this

[1]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

increase can be compensated for by raising the MRT. Similarly, with a higher air temperature the same total heat loss will be maintained by a correspondingly lower MRT.

The loss by evaporation depends on the air temperature, air movement, and humidity; it is increased if the humidity is reduced. For the usual conditions of heating by radiators or convectors, where the air temperature ranges from 70 F to 73 F, approximately 75 per cent of the total heat loss of 400 Btu per hour occurs by radiation and convection, and the balance, or 100 Btu per hour, occurs by evaporation. In the case of radiant heating, if the air temperature is reduced to 60 F, 84 per cent of the 400 Btu per hour, or 336 Btu per hour, is lost by radiation and convection, and 64 Btu per hour are lost by evaporation.

The mean normal surface temperature of the human body, taken over the whole area, including not only the exposed skin surface but also surfaces of the clothes and the hair, has been very extensively used as 75 F, particularly in British literature. However, results obtained by Aldrich[2] in rooms in which the air and wall surface temperatures were approximately 72 F gave mean values nearer to 83 F than to 75 F.

The mean body surface temperature which will maintain the optimum heat loss by radiation and convection in a uniform environment of 72 F may be calculated from fundamental equations for radiation and natural convection by substituting a comparable cylinder for the body. Heilman[3] gives the following equations:

$$H_r = 0.1723\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \qquad (1)$$

$$H_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \qquad (2)$$

where

H_r = heat loss by radiation, Btu per square foot per hour.
H_c = heat loss by convection, Btu per square foot per hour.
T_s = absolute temperature of the body surface, degrees Fahrenheit.
T_w = absolute temperature of the walls, degrees Fahrenheit.
T_a = absolute temperature of the air, degrees Fahrenheit.
$T_m = \frac{T_s + T_a}{2}$
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission.

If it be assumed that a normal adult has an average height of 5 ft 8 in. and an average body surface area of 19.5 sq ft, the surface of his body will have the same area as that of a cylinder 5 ft 8 in. long with a diameter of 13.15 in. The value of e for skin and clothing is practically 0.95. T_a and T_w are each taken as 72 F, or 532 Absolute. The sum of H_r and H_c is taken to be 15.4 Btu per square foot per hour, which is derived as the normal rate of heat loss due to convection and radiation from a sedentary individual by dividing his total sensible heat loss by his area. Solution of

[2]A study of Body Radiation, by L. B. Aldrich (Smithsonian Miscellaneous Collections, Vol. 81, No. 6, December, 1928).

[3]Surface Heat Transmission, by R. H. Heilman (*Trans. A.S.M.E., Fuels and Steam Power Section*, Vol. 51, No. 22, September-December, 1929).

Equations 1 and 2, using average figures as outlined, gives a value of approximately 83 F for the normal temperature of the body surface. This agrees more closely with the values obtained by Aldrich than with the 75 F used by British investigators.

British Equivalent Temperature

The British Equivalent Temperature BET is the temperature of an environment which is effective in controlling the rate of sensible heat loss from a sizable black body in still air when the body has a maintained surface temperature of 83 F. The BET is, therefore, a function of both the air temperature and the mean radiant temperature. Its numerical value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and air. In a non-uniform environment (walls and air at different temperatures) the BET is equivalent to that of a uniform environment in which an 83 F surface loses sensible heat at the same rate as it does in the non-uniform environment. As originally defined, the BET was based on a body surface temperature of 75 F, but 83 F has been accepted as giving results more nearly conforming with American practice[4]. The higher the BET the less the heat loss from the body, the rate of loss in still air being approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no sensible heat loss from a surface at that temperature, so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated.

When convected heat is used, the temperatures of the air and walls are nearly the same, and the optimum value of the BET from the physiological point of view is 72 F. Under these conditions the mean surface temperature of a normal body would have the optimum value of 83 F because the rate of heat loss by radiation and convection would be 15.4 Btu per square foot per hour and that by evaporation 5.1 Btu per square foot per hour, which would just balance the rate of heat production of 20.5 Btu per square foot per hour. This BET of 72 F in a uniform environment is exactly equivalent to the *effective temperature* of 66 F as defined by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (see Chapter 2), because. in a uniform environment, a dry-bulb temperature of 72 F in still air with a relative humidity of 30 per cent gives an *effective temperature* of 66 F, which has been determined to be the optimum.

METHODS OF APPLICATION

There are two general methods of applying radiant heating, as follow:

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls, ceiling or floors, the heating medium being hot water or, in some cases, steam. This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should not exceed about 120 F on account of the possibility of cracking the plaster, the

[4]Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A.S.H.V.E. Journal, *Heating, Piping and Air Conditioning*, July, 1933).

area of the panel must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces comfortable and economical results.

2. *By attaching separate heated plates or panels to the interior surfaces of the structure.* These plates or panels are placed either in an insulated recess flush with the surface of the walls or ceiling or bolted on its face. They may be decorated as desired. As it is difficult to make an invisible joint between the edge of such a plate and the plaster, it is common to use a frame of plaster, wood, metal or composition around the panel. These plates may be placed either on the ceiling or the wall, or in some cases as a margin around the edge of the floor. If floor heating is required the temperature over the whole area should not exceed 70 F.

If the entire warm surface is installed at one end of the room there may be a marked difference between the BET on the two sides of a body in the room. It is usually desirable therefore that the heat be distributed at different points in the room so that no uncomfortable effects will be felt from unequal heating.

PRINCIPLES OF CALCULATION

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine the rate of heat loss from the room by conduction, convection, and radiation when maintained in the desired condition; radiant heating involves the regulation of the rate of heat loss per square foot from the human body.

The first step in the calculations for radiant heating is to ascertain the necessary mean radiant temperature (MRT); next, the size, temperature, and disposition of the heating surfaces required in the room to produce this MRT are estimated; and after this the determination of the convective heat is made.

Mean Radiant Temperature

If the whole of the interior surface of a room were at the same temperature, this temperature would represent the MRT. Such a condition seldom exists, however, since the actual surface temperature in any heated space having surfaces exposed to the outer air varies greatly for different sides of the enclosure. It is therefore necessary to ascertain by calculation the mean of these interior surface temperatures.

The mean temperature in this sense is not the arithmetic average of the actual thermometric temperatures of the surfaces, but the temperature corresponding to the average rate of heat emission per square foot of surface. The temperature corresponding to this mean emission can be taken from Table 1. Conversely, the emission at different temperatures and also the emissivity factors can be obtained from this table. For instance, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero if the emissivity of the surface is 0.9.

If the area in square feet of each part of the space is multiplied by the emission value corresponding to its actual temperature, and these products are added together, the gross amount of radiant heat discharged into the room by the wall surface per hour is obtained. This quantity, divided by the total interior surface, gives the average amount of heat coming into the room from the surface of the walls per square foot of surface per hour.

Interpolating in Table 1, the total radiation from a surface at 83 F for

TABLE 1. TOTAL BLACK BODY RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126.6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula

$$Q = e\left(\frac{0.1723 \times T^4}{100,000,000}\right)$$

where

Q = total black body radiation, Btu per square foot per hour.
e = emissivity.
T = absolute temperature, degrees Fahrenheit.

an emissivity of 0.95 is 142 Btu per square foot per hour. The difference between 142 Btu and the average amount of heat coming into the room is the amount which will be lost per square foot per hour by radiation from a body at 83 F. If a rate at which it is desired that heat be lost from the body by radiation and convection be assumed, the mean radiant emission from the walls required to give the desired result can be determined from Table 1, as can also the required air temperature for the corresponding convective effect.

The determination of the amount of radiant heating surface needed in a room requires knowledge of the climate, the type of structure, the type of heating, and the surface temperature of the walls. This problem can be solved only on an empirical basis. After some experience, however,

it is possible to estimate these variables with a considerable degree of accuracy for any climate or construction.

Assume that a mean radiant temperature of 65 F is desired. Table 1 shows that with all the walls at this temperature, and with an emissivity of 0.95, the gross heat emission is 124 Btu per square foot per hour. The total emission of radiation into the room from that surface would therefore be $A \times 124$, where A is the total inside area of the room. This is the *desired* emission.

If the whole area be divided into a number of different parts which are each at a uniform temperature—a_1, a_2, a_3,—and each is multiplied by the value of the heat emission corresponding to that temperature, and if all these products are added together, their sum will represent the total *actual* emission of radiation into the room at these temperatures without the aid of any hot surface.

The difference between the desired emission and the actual emission represents the additional heat which must be supplied by the hot surface. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. This emission is divided into the additional amount of heat needed, adjusted for the fact that the heating units will shield the walls behind them, and the quotient obtained will be the area of the required heating surface.

It is evident that this method of calculation is approximate, and depends for its accuracy on a correct estimate of the ultimate surface temperatures attained by the actual wall surfaces.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection, and thereby to determine whether this amount of convected heat will warm entering ventilating air to the temperature maintained. If it will not, additional convection surfaces must be introduced to make up the deficiency.

MEASUREMENT OF RADIANT HEATING

Convection heating, having as its object the raising of the air temperature to a specified degree, must be measured by thermometric methods which indicate essentially the air temperature, and not the rate of heat loss from the human body. Radiant heating, having as its object the control of the rate of heat loss from the human body, can be measured only by methods which basically are calorimetric, that is, which measure directly the rate of heat loss from an object maintained at the temperature of the body, irrespective of air temperature.

The apparatus for this purpose consists essentially of a hollow sphere, or cylinder, containing a fluid which can be maintained accurately at 83 F (the accepted mean surface temperature of the human body), with an accurate means of measuring the rate of heat supply required to maintain the temperature at that exact point. The latter measurement can be made with sufficient accuracy by electrical methods. Although a BET of 72 F is desirable, the mean radiant and air temperatures may both vary, provided the heat loss by radiation and convection from a surface at 83 F is maintained at the rate of 15.4 Btu per square foot per hour,

which corresponds to $\frac{15.4}{3.415} = 4.5$ watts per square foot of exposed surface.

This instrument, the *eupatheoscope*, can readily be adapted as a thermostat by electrical control to shut off or turn on heat when the critical temperature of 83 F in the vessel is increased or decreased. A modification of the instrument is called the *eupatheostat*.

Another instrument for maintaining comfort conditions is at present available only in a model adapted to British practice as it is designed for a temperature of 75 F. It consists of a blackened copper sphere of approximately 6 in. diameter in which is housed a cylindrical sump containing a volatile liquid. In operation, a small electric heating coil drawing about 5 watts creates in the sphere a vapor pressure which is constant as long as the heat losses from the sphere are standard. If the temperature of the air or the MRT becomes too high for comfort, a greater pressure is created, owing to a smaller loss of heat from the sphere. This increase of pressure acts on a diaphragm and shuts off the supply of heat to the room.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 in. to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature*.

EXAMPLE

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly representative of American practice with well-built walls, and are based on an emissivity of 0.95 which approximates that of most paints and building materials.

TABLE 2. SURFACE AREAS, TEMPERATURES, AND EMISSIONS FOR A ROOM OF 5760 CU FT

	Area Sq Ft	Assumed Surface Temperature (Deg Fahr)	Heat Emission (Btu Per Sq Ft per Hour)	Total Heat Emission from Area (Btu per Hour)
External Wall	297	50	110.6	32,850
Glass	279	45	106.5	29,710
Inner Wall	480	55	115.1	55,250
Ceiling	480	55	115.1	55,250
Floor	480	55	115.1	55,250
Total	2016			228,310

The mean radiant temperature of the room is $\frac{228{,}310}{2016} = 113.2$ Btu per square foot per hour which, as seen from Table 1, corresponds to an MRT of 53 F for an average emissivity of 0.95.

For an average individual having a body surface of 19.5 sq ft, under conditions of comfort with a body surface temperature of 83 F, the heat given off by radiation may be determined by means of Equation 1 as 217 Btu per hour, or 11.1 Btu per square foot per hour. This corresponds to an environmental emission of $142 - 11.1 = 130.9$ Btu per square foot per hour, and, according to Table 1, to an MRT of 72 F.

If this body be placed in the room described, it will lose heat at the rate of $19.5 (142 - 113.2) = 562$ Btu per hour. This loss is 345 Btu per hour, or 17.7 Btu per square foot per hour, more than the rate of heat loss for comfort, which is only $19.5 (142 - 130.9) = 217$ Btu per hour.

In order to determine the amount of radiating surface necessary to maintain the MRT at 72 F, assume the surface temperature of the hot plates to be installed to be 200 F, which is approximately the temperature they would have if heated by steam.

The 2016 sq ft total area of the surfaces of the room multiplied by 130.9, which is the emission in Btu per square foot per hour necessary to maintain a body surface temperature of 83 F, gives a total desired emission of 263,890 Btu per hour. It is necessary to supply enough radiant heating surface to increase the total actual mean radiant heat emission by the room from 228,310, as shown in Table 2, to the 263,890 Btu desired. The additional heat needed is the difference between these figures, or 35,580 Btu. Since, from Table 1, the emission per square foot at 200 F is 309 Btu, the required radiant heating surface needed is $\frac{35,580}{309} = 115$ square feet. The effect of this surface suitably placed would be to raise immediately the mean radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

In the solution of this particular example, the radiation loss from the human body was selected as 217 Btu per hour, which is that taking place under optimum comfort conditions, with a body surface temperature of 83 F in a uniform environment at 72 F. The mean radiant temperature necessarily was 72 F. If the optimum BET of 72 deg Fahr is desired, an air temperature of 72 F also must be maintained. If it is desired to maintain a lower air temperature than this, a mean radiant temperature greater than 72 F must be selected and the radiation loss from the individual must be recalculated from Equation 1.

The calculation may be simplified by preparing tables showing, at the usual temperatures, the area of hot surface required to bring each square foot of actual wall surface at various temperatures up to a general standard of from 60 F to 70 F. It would then be necessary only to multiply the respective areas by the appropriate factors, and to add the results, to obtain the required total.

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PROBLEMS IN PRACTICE

1 ● Name three ways that heat is lost from the human body.

By radiation, convection, and evaporation.

2 ● What is the mean normal surface temperature of the human body as determined for the United States?

83 F.

3 ● What is the exact purpose of radiant heating?

Radiant heating regulates the heat loss from the human body.

4 ● How is the required amount of radiant heating surface found?

By calculating the *desired emission* and the *actual emission*, and finding their difference. This is additional heat which must be supplied by the hot surface.

5 ● After finding the required heat, how is the necessary hot surface area calculated?

Having selected the hot surface temperature, find the emission per square foot from Table 1. This rate divided into the heat required gives the area of the necessary heating surface.

6 ● Is the heat generated in the body affected by action? If so, does it vary greatly?

Yes. With hard work or energetic exercise, the total heat generated in the body may be five to six times that generated when it is at rest.

7 ● When and why does the human body feel cold?

The body feels cold not only when it loses heat at a greater rate than it can generate it, but also when heat is abstracted from the body disproportionately. The human body does not require any heat from without because it generates more heat than is sufficient to maintain the correct temperature; therefore, it is only necessary to provide conditions that will maintain the correct ratio of losses.

8 ● a. Where did radiant heating derive its name?
b. What is actually meant by radiant heating?

a. The term radiant heaters was introduced about 25 years ago to designate flat heating surfaces made to give off practically all their heat by radiant ether waves instead of relying on convected warm air.

b. The term radiant heating now applies to methods of heating where, instead of heating the air to a predetermined temperature, flat heating surfaces are so placed in a room that the average virtual temperature of all wall, ceiling, floor, and glass surfaces exposed to the body is just sufficient to prevent the body's losing too much heat by radiation. The air temperature can be much cooler with radiant heating because radiation losses from the body are compensated.

9 ● What kind of heating surfaces are in general use?

The heating units may have flat iron surfaces heated with steam and placed under windows, or hot water pipes may be embedded in the floor, walls, or ceilings. Electrical radiant heaters are made by embedding resistance elements in porcelain or electric conductors woven into a thick paper which can be fastened to the walls or ceilings.

10 ● What kind of heat rays are commonly generated in radiant heating? Give examples.

All heat rays are generally assumed to be the same as light rays; they travel at the speed of light, but they are invisible and longer. The rays used in heating are 0.00005 to 0.0001 in. long compared with visible red rays of about 0.000027 in.

11 ● What natural evidence have we that air temperature alone is no criterion of comfort and that radiant heat affects the body more quickly?

When standing in the sunshine on a cool spring day, a person feels perfectly comfortable, but when a cloud passes over the sun, he instantly feels much cooler as the shadow reaches him. A shielded thermometer recording the temperature of the air shows no reduction in air temperature in so short a period, so that the person actually feels a sensation of cold which an ordinary thermometer cannot register. This shows that light and heat rays are shut off simultaneously and travel at the same speed; it also proves that radiant rays affect the comfort of the body quicker than air temperature does.

Chapter 39

ELECTRICAL HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Steam Heating, Electric Hot Water Heating, Heat Pump, Control, Calculating Capacities, Power Problems, Electric Heating Data

ELECTRIC heating has a logical and a rapidly growing place in the heating industry because of its advantages of flexibility, cleanliness, safety, convenience, and ease of control. Electric heating practice has many basic principles in common with fuel heating, but there are also important differences. The advantages of good building insulation are even more important in electric heating than for fuel heating, because the initial cost per Btu is usually higher.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, because 100 per cent of the energy applied in the resistor is always transformed into heat. In electric heating practice the engineer need not be concerned about efficiencies of heat production, but rather about efficiencies of heat utilization.

DEFINITIONS

Definitions of terms used in fuel heating are given in Chapter 41. The following terms apply particularly to electric heating:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common

resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

HEATING ELEMENTS

Commercial electric heating elements are divided into open type elements, enclosed type elements, and cloth fabrics. *Open type elements* have resistors exposed to view. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. Open type elements are used extensively for operation at high temperatures when radiant heat is desired. They are also frequently used at low temperatures for convection and fan circulation heating, especially in large installations.

Enclosed type elements have metallic resistors embedded in a refractory insulating material, and encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Enclosed elements are made in many forms, such as strips, rings, plates, and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in convection and fan circulation air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. *Cloth fabrics* woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes.

ELECTRIC HEATERS

Electric heaters are classified according to the manner in which they deliver heat in practical use, that is, by conduction, by radiation, or by convection. The term *radiator* should not be used in electric heating, because of confusion between its established usage in fuel heating and the radiant principle of many electric heaters.

Among the uses of *conduction electric heaters*, which deliver most of their heat by actual contact with the object to be heated, are aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and pipe heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature incandescent heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. The location of radiant heaters is important. They should never face a window because some rays would pass through the glass and be lost. Figs. 1 and 2 show common types of portable and wall-mounted radiant heaters.

Convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in

much the same manner as steam and hot water radiators or convectors. They should have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line (Figs. 3 and 4). The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction.

UNIT HEATERS

Fan unit electric heaters, having electric heating elements combined in the same enclosure with a fan or blower, are made in many styles and are excellent for general air heating. They should be located and used much

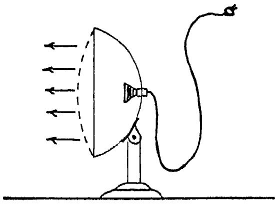


FIG. 1. PORTABLE RADIANT ELECTRIC HEATER

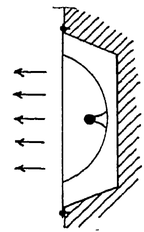


FIG. 2. RADIANT ELECTRIC HEATER RECESSED IN WALL

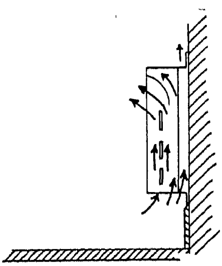


FIG. 3. CONVECTION ELECTRIC HEATER ON WALL SURFACE

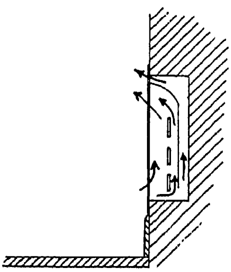


FIG. 4. CONVECTION ELECTRIC HEATER RECESSED IN WALL

as steam unit heaters. The warm air can be directed toward the floor, if desired, to give a positive circulation which will reduce stratification of air. Small units which are free from radio interference are used for homes; there are large units for industrial plants, substations, power houses, and pumping stations; portable units are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction (Figs. 5, 6, 7 and 8).

CENTRAL FAN HEATING

Central fan electric heating systems have electric heating elements and fans or blowers to circulate the air through ducts, and in addition to the

main heaters at the fan location, booster heaters may be located in branch ducts. Humidification or complete air conditioning can readily be included in the system, in much the same manner as with steam.

In coördinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure as a change in air volume flowing over steam coils does not greatly affect the temperature of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains

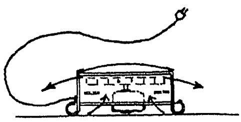


FIG. 5. SMALL PORTABLE FAN UNIT ELECTRIC HEATER

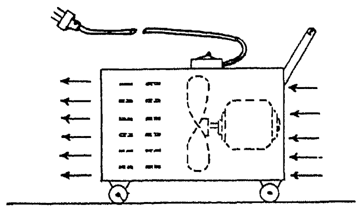


FIG. 6. LARGE INDUSTRIAL TYPE PORTABLE FAN UNIT ELECTRIC HEATER

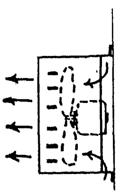


FIG. 7. SMALL FAN UNIT ELECTRIC HEATER

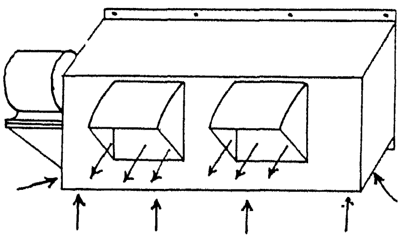


FIG. 8. LARGE INDUSTRIAL TYPE FAN UNIT ELECTRIC HEATER

about the same. Electric heat is quite different, being a constant source of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic modulation to vary the electrical heat input and synchronize it properly with the air flow has been successfully applied to central fan systems.

ELECTRIC STEAM HEATING

Electric steam heating differs from fuel heating only in the use of *electric boilers* to generate steam. Small boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used on either direct or alternating current since the

heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood that the heating elements will burn out, they are made removable for cleaning off deposits of scale which will restrict the heat flow. Boilers of this type are useful in industrial plants which require limited amounts of steam for local processes, and for sterilizers, jacketed vessels, and pressing machines which need a ready supply of steam.

Electric boilers are entirely automatic and are well adapted to intermittent operation. It frequently is economical to shut down the main plant boilers when the heating season ends, and to supply steam for summer needs with small electric boilers located close to the operation. Large electric boilers are usually of the type employing water as the resistor. Only alternating current can be used, as direct current would cause electrolytic deterioration. Large boilers of this kind have electrodes immersed in the water where heat is generated directly. In Canada and Europe many successful installations have been made, but in the United States the cost of electric power, in comparison with fuels, does not favor its general use.

ELECTRIC HOT WATER HEATING

Electric hot water heating offers an extremely convenient and reliable means of supplying all needs for hot water, and in sections of the country where low current rates have made it economically feasible, it enjoys popularity. Electric boilers for hot water heating are inexpensive, entirely automatic, and are insulated to prevent excessive heat losses. When lower power costs can be secured, by confining the heating to certain fixed hours, water may be heated and stored in well-insulated tanks for use when needed. In large industrial plants it is often possible to balance power loads by this means and to avoid running the fuel-fired steam boilers at night or over week ends. In Europe use has been made of this hot water storage principle for heating. Experiments have been made in this country for heating houses, but the cost of serving individual homes with the necessary heavy electric power loads has proved unprofitable at rates comparable to other forms of heating. The problems incident to installing large storage tanks in home basements, and the lack of flexibility under variable weather conditions, are also unfavorable factors.

OIL HEATING

Electric hot oil heating is useful in some industrial work as a substitute for superheated steam. Special oil can be electrically heated as high as 600 F and pumped at a pressure just sufficient to cause flow. When used in heating coils or jacketed vessels, this gives a safe, and convenient, automatic system for moderate-sized installations.

HEAT PUMP

The electric heat pump is not strictly an electric heater, as it does not directly convert electrical power into heat. It operates a compressor electrically which acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and

discharge it outdoors. This system has been used in evenly-balanced climates where the heating requirements in winter are about the same as the cooling requirements in summer.

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electrical heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a house or building comfortable. Likewise, such electrical heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down are conveniently cared for electrically.

CONTROL

Because the efficiency of electric heat production is the same for large or small units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Wherever radiant heaters are used, thermostats are not an effective means of control and manual operation or control by eupatheoscope is necessary. For all convection and fan circulation heaters thermostatic control is useful. For small heaters having ratings up to about 1500 watts, there are direct-acting thermostats which are satisfactory, but for larger heaters it is advisable to use relays or contactors, which should break all of the power lines. All heaters having fan circulation should have the heat circuit interlocked with the motor circuit so that the fan will be running when the heat is on. A thermal fuse or trip should be located in the heat chamber to throw off the heat in case any interruption of air flow should occur; otherwise undue temperature rise would result. In all large heaters the heating elements should be arranged in groups and control provided to vary the heat input to correspond approximately to the heat demand. If this is not done, and all the heat is kept available, the thermostat will continue throwing it on and off at short intervals. Except for central fan systems, the heat stages can be operated by manual switches, but automatic modulation of the heat load is usually preferred.

CALCULATING CAPACITIES

The methods of calculating heat losses outlined in Chapters 6, 7, and 8 may be used for electric heating exactly as for fuel heating. The total heat requirements in Btu per hour may then be converted into the electrical rating of an equivalent heating system by using the equation:

$$\frac{\text{Total Btu per hour}}{3415} = \text{kw rating of required electric heating} \qquad (1)$$

The following empirical rules for estimating electric heater require-

ments may be used in territories where the heating load is never greater than 1500 degree days:

Capacity of heater required for average room in home..............2 watts per cubic foot.
Capacity of heater required for average office occupied
in the daytime only..1.2 watts per cubic foot.

POWER PROBLEMS

The first point to determine is the cost of the power which is available for electric heating. Unlike fuels, there is no uniform cost for electric power because of the unequal cost of distribution to large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate power plants at uniform loads; hence, even the time of use may affect the cost of power.

Homes are almost universally supplied with lighting current of 115 volts, which cannot be used economically for any but the smallest heaters. Usually the service lines will not permit more than plug-in devices. The underwriters permit heaters of 1250 watts to be used from approved baseboard receptacles. Where homes have 230 volt service for cooking and water heating, and rates are favorable, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 230, 460 or 575 volts. All polyphase heaters should be balanced between phases.

ELECTRIC HEATING DATA

Electric heater capacity is rated in kilowatts (kw). Electric energy is measured in kilowatt-hours (kwhr). Cost of operation = kw rating × hours used × cost per kwhr.

One boiler horsepower (bhp) = 33,471.9 Btu per hour.
One kilowatt-hour (kwhr) = 3,415 Btu.

$$\text{One boiler horsepower} = \frac{33{,}471.9}{3{,}415} = 9.80 \text{ kwhr.}$$

One boiler horsepower will evaporate 34.5 lb water per hour *from and at* 212 F.

$$\text{One kilowatt-hour} = \frac{34.5}{9.80} = 3.52 \text{ lb of water per hour at 212 F.}$$

Additional conversion factors are given in Chapter 41.

PROBLEMS IN PRACTICE

1 ● Why is electrical energy economically feasible to use for certain heating applications?

a. Because heat from the radiant type of electrical heater is effective in producing comfort for the occupant almost as soon as it is turned on, the heater may be turned off when the room in unoccupied.

b. There is nothing to freeze in an electrical heater.

c. Electrical energy may be purchased at lower rates during off-peak periods and stored as heat in water kept in insulated tanks until needed.

d. There is no wasted energy up the flue or in the ash as with fuel heating.

2 ● To what localities is electrical heating adaptable?

To those localities where the heating season is relatively mild and where electrical energy is available at low cost, as in communities served by large hydro-electric plants.

3 ● Approximately how low must the rates be to permit the use of electricity for heating purposes?

Probably the energy must sell for 2 cents or less per kwhr. At 2 cents the cost would be $5.86 per 1000 Mbh. (See Chapter 29 for comparison with other fuels.) This looks high, but the seasonal energy consumption would not be as large with electricity as with other fuels, for reasons stated in Question 1.

4 ● Why is automatic control important in connection with electrical heating?

The higher cost of the energy makes it essential that none be wasted.

5 ● In fan heating systems, what is an important difference between a steam heated coil and an electrically heated coil?

A coil supplied with steam at constant pressure will remain at constant temperature regardless of the amount of air passing over it. The temperature of the electric coil supplied with a constant amount of energy will rise if the air quantity is decreased and fall if the air quantity is increased.

Chapter 40

TEST METHODS AND INSTRUMENTS

Pressure Measurement, Temperature Measurement, Air Movement, Humidity Measurement, Carbon Dioxide Determination, Dust Determination, Flue Gas Analysis, Measurement of Smoke Density, Heat Transmission, Eupatheoscope

ATMOSPHERIC pressure is usually measured by a *mercurial barometer* which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The pressure of the atmosphere on the exposed top of the mercury in the cistern supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern. Atmospheric pressure is the same as the pressure exerted by this supported column of mercury, and, in pounds per square inch, is equal to its height in inches times 0.491, which is the weight in pounds of 1 cu in. of mercury. At latitude 45 deg and sea level, and at a temperature of 32 F, the atmosphere will support a column of mercury 29.921 in. in height. The pressure of 14.7 lb per square inch, derived by multiplying 29.921 by 0.491, is called *standard* or *normal barometric pressure.* Since the height of the barometer depends on the density of the mercury as well as on the pressure of the atmosphere, and since the density is dependent on the temperature, mercurial barometer readings should always be corrected for temperature. An *aneroid barometer* contains no liquid; it is portable but less accurate than the mercurial barometer. Atmospheric pressure in bending the thin corrugated top of a partially exhausted metallic box, or in distorting a thin-walled bent tube of metal, is made to move a pointer.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage,* and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage.* The most common type of these gages contains a flexible hollow brass tube of oval cross section, known as a *Bourdon tube.* When subjected to unequal inside and outside pressures, this tube tends to straighten out, and a pointer motivated by this straightening indicates the pressure difference on a suitably graduated scale.

High vacuum readings such as are encountered in condenser and steam jet refrigeration practice are commonly obtained by the use of mercury column vacuum gages. When the readings obtained with the mercurial barometer and those with the mercury vacuum gage have both been corrected to 32 F, the difference in the two readings will give the absolute

vacuum in inches of mercury. The following equation may be used to make corrections for temperature:

$$h = h_1 [1 - 0.000101 (t_1 - t)]$$

where

h = height of mercury column corrected to temperature t.
h_1 = actual height of mercury column.
t_1 = actual temperature of mercury column.
t = temperature to which column is to be corrected.

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage*. It is essentially a *U* tube containing either water, kerosene, alcohol, or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pressure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

TEMPERATURE MEASUREMENT

In engineering work, *mercurial thermometers* are largely employed to measure the intensity of heat. These depend on the uniform expansion of mercury to indicate changes in temperature. An amount of mercury held in a sealed tube with a bulb at one end will rise to one definite level when immersed in melting ice, and to another definite level when immersed in boiling water. These two points are marked, and the space between them is divided into a number of equal portions, each of which is called a degree. In the Fahrenheit scale, there are 180 degrees thus obtained, while the centigrade scale has 100 and the Réaumur has 80. Like divisions are marked off on the column above and below these two determined points in order that a greater range of temperature may be read.

Thermocouples[1] may be used to measure any range of temperatures up to 2,900 F. When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends on the composition of the wires and the difference in temperature between the junctions. A potentiometer or sensitive galvanometer of high resistance connected to the thermocouple will give a deflection which is a function of the temperature difference between the hot and cold junctions. Thermocouples connected in series are called *thermopiles*. Thermocouples for the measurement of high temperatures are calibrated with the aid of the known melting points of pure metals.

Resistance thermometers are suitable for temperature measurements up to 1800 F. These thermometers depend for their operation on the change of resistance with temperature of a platinum, nickel, or copper wire coil, and they are calibrated in the same way as thermocouples.

[1]See A.S.H.V.E. research paper entitled Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

For temperatures above 500 F various types of *pyrometers* are employed. The *mercurial pyrometer* is a thermometer with an inert gas, such as nitrogen or carbon dioxide, above the mercury column to prevent the mercury from boiling. The *radiation pyrometer* consists of a thermopile upon which the radiation from a hot source is focused by a concave mirror. A sensitive galvanometer with a calibrated temperature scale indicates the thermo-electromotive force created by the heat on the thermopile. The *optical pyrometer* measures radiant energy by comparing the intensity of a narrow spectral band, usually red light emitted by the object, with that emitted by a standard light source (electric lamp). *Thermo-electric pyrometers* operate on the same principle as thermocouples. When measuring high temperatures, it is customary to hold the cold junction at room temperature and this may cause some error if the room temperature is above or below the calibration point. For extremely precise temperature measurements, the cold junction is usually immersed in melting ice to fix the cold junction temperature. Various forms of hand-operated and automatic cold junction temperature compensators are also available.

In the measuring of room temperatures care must be exercised to prevent the results from being affected by the body heat of the observer, by drafts from doors, windows and other openings, or by radiant heat from some local source such as a radiator or wall. All thermometers should be mercury thermometers with engraved stems. The total graduations of the thermometers should be from 20 to 120 F, in one degree graduations. No ten degrees should occupy a space of less than one-half inch. The accuracy throughout the whole scale must be within one-half degree. The operator should take hold of the top and no part of the body, including the hand, should be nearer than 10 in. to the bulb. The thermometer should not be closer than 5 ft to any door, window, or other opening; should not be closer than 12 in. to any wall; and should be between 3 and 5 ft from the floor. A sling instrument should be used for extreme accuracy. Thermocouples or resistance thermometers may also be used for room temperature measurements, an advantage being that the operator can read temperatures from outside the room if desired, and thus eliminate the errors which might be caused by his presence close to the temperature measuring device.

For measuring duct temperatures a duct thermometer should be used, with the bulb extending into the duct at least 6 in. When the thermometer is to be permanently located in the duct, a pipe flange or nipple should be used to receive the threaded portion of the thermometer stem. When the thermometer is not to be permanently located, a cork or rubber stopper may be placed around the stem to prevent errors from air leakage. Readings should be taken at various locations in a duct so due consideration may be given to temperature stratification. Other forms of temperature measuring devices may be used, but the active part must be at least 6 inches from the duct wall.

Recording thermometers may be used for testing, and for making continuous records of operation. Care should be taken, however, to insure that time lag due to heavy measuring elements is kept to a minimum, so that the recorders will properly follow temperature fluctuations. Thermocouples made of fine wire will show less time lag than will many mercury bulb thermometers.

MEASUREMENT OF AIR MOVEMENT

The quantity, velocity and pressure of air moved by a fan or flowing through a duct or grille may be determined by various methods. The instruments in common use are the Pitot tube, anemometer, direct reading velocity meter, and Kata-thermometer, the latter being suitable for low air velocities and being commonly used for measurements at points where the air is not confined in a duct. The use of calibrated nozzles, orifice plates, and Venturi meters are recognized methods, which, however, have little application in connection with ventilation practice.

Pitot Tube

This usually consists of two tubes, one within the other, which when properly held in the air stream will register the total or impact pressure and the static pressure, respectively. If these tubes are connected to opposite sides of a water column, or other type of manometer, the recorded pressure will be the differential or velocity pressure. Volume measurements may thus be made in a duct of known area. Pitot tube measurements are preferably used for air velocities exceeding 20 fps. Volumetric determinations from Pitot tube readings should take into account the barometric pressure and the temperature and humidity of the air measured.

In general no accurate velocity pressure readings can be taken when the flow of air in ducts is turbulent. To insure accuracy a straight section of duct from 5 to 10 times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work it is necessary to make a traverse of the duct, dividing its cross section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

Anemometer

This instrument is delicate, and requires frequent calibration when accuracy is desired. The vanes of the instrument should never be touched and it should never be held in air having a velocity greater than that for which it is calibrated. Readings taken directly in a fan inlet or discharge are likely to harm the instrument because of excessive velocities. In duct measurements the same procedure is followed as for the Pitot tube. The anemometer usually reads directly in linear feet. To obtain the velocity in feet per minute, the reading must be divided by the elapsed time in minutes.

The following procedure for obtaining anemometer readings is based on research conducted at *Armour Institute of Technology* in coöperation with the A.S.H.V.E. Research Laboratory[2].

Supply Grilles. The surface of the grille should be marked off into a number of equal areas approximately 6 in. square. A 4-in. anemometer

[2]Measurement of Flow of Air through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, Vol. 37, 1931, and A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, September, 1933).

should be used and should be held at the center of each section in contact with the grille (or as close as possible) for a period of time sufficient to insure an average reading. In the case of supply grilles, the instrument should always be held with the dial facing the operator. The average of the corrected readings should then be used in the following formula to obtain the flow in cubic feet per minute:

$$cfm = CV\frac{A + a}{2} \text{ or } \frac{CVA\,(1 + p)}{2} \qquad (1)$$

where

V = average of corrected anemometer readings, feet per minute.
A = gross area of grille, square feet.
a = net free area of grille, square feet.
p = percentage of free area of grille expressed as a decimal.
C = a coefficient that varies with the velocity from grille and may vary slightly with type of grille. For average use, with supply grilles, C can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities.

Particular care should be exercised in the case of long, narrow grilles. The nature of the approach sometimes results in there being a narrow strip along the top or bottom of the grille through which no air will be flowing. This may be detected by holding the anemometer completely out of the air stream and then moving it slowly inward over the grille until the vanes just start to move. The distance which the vanes extend over the grille opening at this moment will indicate the width of the dead strip. Only the remaining portion of the grille should be considered in making the calculations for gross and free area.

Exhaust Grilles. The surface of the grille should be marked off and readings taken in the same manner as with supply grilles, except that the instrument should be held with the dial facing the grille, and in contact with it. The traverse should be taken at a uniform rate, allowing sufficient time in each space to minimize the percentage of error. In the case of exhaust grilles it is found that the formula

$$cfm = KVA \qquad (2)$$

in which

V = average indicated velocity obtained by the anemometer traverse.
A = gross area of grille, square feet.
K = coefficient determined by experiment. For average use, with exhaust grilles, K may be taken as 0.8 for all usual velocities.

This formula is of advantage, especially with ornamental grilles, in that the free area need not be measured.

The flow of air through registers and grilles is of considerable importance, being frequently the only convenient method of measuring the volume of supply air to a room. While duct measurements, if available, are more dependable, grille measurements provide a fairly accurate method, if care is taken in the technique of using the anemometer.

Kata-Thermometer

The Kata-thermometer can be used to determine air velocities provided the walls and surrounding objects are at or near the room tem-

perature. Especially at low velocities it constitutes a useful instrument for readily detecting drafts.

The instrument is essentially an alcohol thermometer with a bulb approximately $\frac{5}{8}$ in. in diameter and $\frac{1}{2}$ in. long with a stem 8 in. long reading from 100 F to 95 F, graduated to tenths of a degree. To take readings the bulb is heated in water until the alcohol expands and rises into a top reservoir. The time in seconds required for the liquid to fall from 100 F to 95 F is recorded with a stop watch and this time is a measure of the rate of cooling.

The dry Kata loses its heat by radiation and by convection so for constant velocities the time of cooling is a function of the dry-bulb temperature of the surrounding air. The wet Kata, which has a cloth covering fitted snugly around its bulb, loses heat by radiation, convection, and evaporation, and for constant velocities its rate of cooling is a function of the wet-bulb temperature of the air irrespective of the dry-bulb temperature or relative humidity. It does not follow, however, that the difference in rate of cooling of the dry and the wet Kata is caused by evaporation. A change in the wet-bulb temperature produces a change in the surface temperature of the wet Kata which in turn affects the heat lost by radiation and by convection.

Several precautions should be taken to obtain the best results with this instrument:

1. To obtain velocity readings use the dry Kata since the error in timing is reduced.
2. The instrument should be heated and allowed to cool two or three times before recording the final time of cooling. The first reading is not reliable.
3. All traces of moisture must be removed from the dry Kata before timing to eliminate error introduced by evaporation.
4. Use only the formula applying to a particular instrument. Each Kata receives an individual calibration.

HUMIDITY MEASUREMENT

The sling psychrometer is the recognized standard instrument for determining humidities. In order to obtain accurate readings considerable skill is required on the part of the operator. The wicking and water must be clean and the temperature of the water should be slightly above the wet-bulb temperature of the surrounding air. The psychrometer should be swung rapidly and several and frequent observations should be made to see that the wet-bulb temperature has become stationary before the final reading is noted. Care should be taken that the wet-bulb has reached a minimum temperature, but the wick must still be moist. Standard psychrometric tables should be used.

In making wet-bulb measurements below 32 F the same procedure is followed as above 32 F. The water is liquid at the start, but as the sling is operated it will freeze rapidly enough so that in quickly giving up the latent heat of fusion, the indicated wet-bulb temperature may drop below the actual wet-bulb temperature. After the liquid on the bulb has become thoroughly frozen the wet-bulb temperature will rise to normal. A very thin film of ice is more desirable than a thick film. Care must be taken to read the temperatures in the region below 32 F accurately because the spread between the wet- and dry-bulb is small.

In taking humidity readings in ducts it is usually impracticable to use a sling psychrometer. For this work the stationary hygrodeik arranged for bolting on to the side of the duct, with two bulbs extending into the duct, will be found very convenient. Owing to the velocity of the air passing over the bulbs within the duct an accurate reading will be secured, corresponding to that given by the sling psychrometer.

Various forms of humidity recorders are available, some merely recording wet- and dry-bulb temperatures, and others recording relative humidity directly. Any form of wet- and dry-bulb device must have sufficient air velocity over the thermometer bulbs to insure accurate readings; this velocity should be secured by a fan if the air is not itself in motion, as in a duct. For extremely low humidities, or for humidity measurements above 212 F, a thermal conductivity method is available[3].

CARBON DIOXIDE DETERMINATION[4]

At ordinary concentrations carbon dioxide is not harmful. The amount of carbon dioxide in the air is a convenient index of the rate of air supply, and of the distribution of the air within rooms. Unequal carbon dioxide concentrations in parts of a room indicate improper air distribution.

The Petterson-Palmquist apparatus has been generally accepted as the standard device for the determination of carbon dioxide in air investigations. The principle involved is the measurement of à given volume of air, the absorption of the contained carbon dioxide in a caustic potash solution, and the remeasurement of the volume of air at the original pressure in a finely graduated capillary tube, the difference in volume representing the absorbed carbon dioxide. (See Report of Committee on Standard Methods for Examination of Air, *American Public Health Association*, Vol. 7, No. 1; *American Journal of Public Health*, Jan., 1917.)

Where field conditions are such that this apparatus may not be conveniently used, as in street cars, air samples may be collected in clean bottles having mercury-sealed rubber stoppers, and these may be subjected to laboratory analysis.

DUST DETERMINATION

Many laboratory methods have been developed to measure the dust in the air. These involve the collection of dust on sticky plates, on filter paper, in water, on porous crucibles, or by electric precipitation, and the subsequent determination of the amount of dust by microscopic counting, weighing, or titration. While there is no standard method, the Hill dust counter, using a microscope, the impinger[5], using chemical changes in water, and the Lewis sampling tube[6], involving the analytical weighing of a porous crucible, are accepted. All test results should be accompanied by the name of the instrument used as great variation in counts with the

[3]"Gas Analysis by Measurement of Thermal Conductivity," H. A. Daynes, Cambridge Press, 1933.

[4]See A.S.H.V.E. research paper entitled Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933).

[5]*Public Health Bulletin, No.* 144, 1925, U. S. Public Health Service.

[6]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, May, 1933).

different instruments will be obtained. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has developed a code[7] for the testing and rating of air cleaning devices used in general ventilation work.

FLUE GAS ANALYSIS

The analysis of flue gases by chemical means is made with the *Orsat apparatus*. A solution of KOH is used to absorb the CO_2. Free oxygen is absorbed by a mixture of pyrogallic acid and KOH. The solution for absorbing the CO is cuprous chloride. The apparatus consists of a burette surrounded by a water jacket, to receive and measure the volume of gas. The burette is connected by a manifold of glass to *pipettes* containing liquids for absorbing CO_2, O_2 and CO.

Various forms of automatic indicating and recording gas analysis devices are available, operating on either chemical or physical principles. Such devices are convenient for plant operation.

MEASUREMENT OF SMOKE DENSITY

Relative smoke density is usually measured by comparison with the Ringelmann Chart (Fig. 1). In making observations of the smoke issuing from a chimney, four cards ruled like those in Fig. 1, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point 50 ft from the observer and conveniently in line with the chimney. At this distance, the lines become invisible, and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of the smoke, and makes a record accordingly, noting the time. Observations are made continuously during one minute, and the estimated average density during that minute recorded. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the entire record is plotted on cross-section paper in order to show how the smoke varied in density from time to time.

Smoke Recorders

Smoke recorders are available which give a much more accurate indication of the amount of smoke being produced than does the Ringelmann Chart. Although most of these instruments are in the process of development, they constitute a satisfactory tool in the control of smoke emission. They all depend upon projecting a beam of light through the smoke flue or through a separate compartment from which a sample of the flue gas is drawn continuously. The light of the beam which passes through without being absorbed by the smoke is measured to determine the smoke density. Most of these instruments make use of a photoelectric cell or a thermopile to measure the relative amount of light which has not been absorbed. Standard electrical instruments serve for indicating or recording.

[7]See A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, edition of July, 1934.

MEASUREMENT OF RATE OF HEAT TRANSMISSION

The standard methods of testing built-up wall sections are by means of the *guarded hot-box*[8] and the *guarded hot-plate*[9]. The *Nicholls heat-flow meter*[9] may be used for testing actual walls of buildings.

It would be obviously impossible to determine the air-to-air heat transmission coefficients of every type of wall construction in use with the heat-flow meter, the guarded hot-box or the guarded hot-plate on account of the great amount of time involved. Hence, the method of computing the coefficients from the fundamental constants must be resorted to in most cases. The guarded hot-plate is used to determine the fundamental

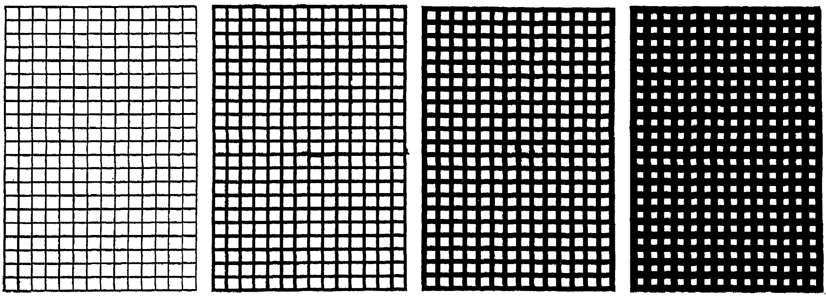


FIG. 1. RINGELMANN SMOKE CHART

constants. The heat-flow meter, guarded hot-box and guarded hot-plate tests can be used to good advantage in checking the accuracy of the computed values.

If the hot-box or hot-plate methods are used, tests are usually run under still air conditions, which means there is no wind movement over the surfaces of the wall during the test. In the hot-plate method of test the inside surface coefficient is eliminated by the plate's being in direct contact with the wall. In practice, some wind movement over the exterior surface of the wall should always be allowed for; hence, still-air coefficients cannot be used over the outside of the building during the heating season. Moreover, still-air transmission coefficients cannot be corrected to provide for moving-air conditions by applying a single constant factor. Computed coefficients of transmission for various types of construction are given in Chapter 5.

EUPATHEOSCOPE

The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort. See Chapter 38.

[8]See Standard Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928) and Report of the Committee on Heat Transmission, *National Research Council.*

[9]See Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).

PROBLEMS IN PRACTICE

1 ● The hand on a pressure gage attached to a steam line indicates a pressure of 15 lb per square inch and the barometric pressure is 14.7 lb per square inch. What is the absolute pressure, in pounds per square inch, being exerted by the steam?

The absolute pressure exerted by the steam in the pipe is equal to the pressure indicated by the gage plus that exerted by the atmosphere.

Total pressure $= 15 + 14.7 = 29.7$ lb per square inch.

2 ● Outline the procedure to be followed in taking room temperatures.

In taking room temperatures, a standard mercury thermometer should be used, with care taken that no part of the observer's body is nearer than 10 in. to the thermometer bulb. The thermometer should be held at least 5 ft away from any window, door or opening; it should be at least 12 in. away from any wall, and should be between 3 and 5 ft from the floor.

3 ● What advantages other than its sensitiveness, has the U-tube draft gage or manometer for measurement of low pressures?

Inherent accuracy without calibration and low cost of the essential parts, which are glass tubing and an ordinary scale.

4 ● Are thermocouples as accurate as mercury thermometers?

Within the range which can be measured with both instruments (below 1000 F) either one may be made as sensitive as the service requires. The accuracy of a thermocouple temperature measurement depends chiefly on:

1. An accurate calibration of the wire.
2. The sensitiveness of the electrical instrument.
3. Accurate cold-junction control.
4. Proper placement of the sensitive junction.

5 ● Is room temperature accurately measured by the ordinary wall thermometer?

No. Wall thermometer measurements may be several degrees in error as compared with an observation properly made in the zone of occupancy.

6 ● When an anemometer is used for measuring the air discharged from a grille or register, does it read the velocity through the gross face area or the velocity through the net free area?

Neither. If either of these velocities is required, it should be calculated by means of Equation 1.

7 ● Do common errors made in humidity determination produce a result that is too high or too low?

A higher relative humidity than the true value is likely to be found, either because there is insufficient velocity over the wet-bulb or because the reading is not taken at the right time.

8 ● What is the purpose of the carbon dioxide determination?

It is an index of the adequacy of fresh air supply and also an indicator of air distribution.

Chapter 41

TERMINOLOGY

Glossary of Physical and Heating and Ventilating Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, A.S.H.V.E. Codes

Absolute Humidity: See *Humidity.*

Absolute Pressure: The sum, at any particular time, of the gage pressure and the atmospheric pressure.

Absolute Temperature: The temperature of a substance measured above *absolute zero.*

Absolute Zero: The temperature (−459.6 F) at which the molecular motion of a substance theoretically ceases. This is the temperature at which the substance theoretically contains no heat energy.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second per second.

$$a = \frac{V}{t}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body. In the fps system this is 32.174 feet per second per second.

Adiabatic: An adjective pertaining to or designating variations in volume or pressure not accompanied by gain or loss of heat. When a substance undergoes adiabatic expansion, since it does not receive heat from without, the work which it does is at the expense of its internal energy, and therefore its temperature falls; similarly, when it is adiabatically compressed its temperature rises.

Adsorption: The adhesion of the molecules of gases or dissolved substances to the surfaces of solid bodies, resulting in a concentration of the gas or solution at the place of contact.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Air Infiltration: The inleakage of air through cracks and crevices, and through doors, windows and other openings, caused by wind pressure or temperature difference.

Air Washer: An enclosure in which air is forced through a spray of water in order to cleanse, humidify, or dehumidify the air.

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure exerted by the atmosphere in all directions, as indicated by a barometer. *Standard atmospheric pressure* is considered to be 14.7 lb per square inch, which is equivalent to 29.92 in. of mercury.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of 970.2 × 34.5 = 33,471.9 Btu per hour.

British Thermal Unit: The *mean* British thermal unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F. It is substantially equal to the quantity of heat required to raise 1 lb of water from 63 F to 64 F. One Btu = $\frac{1}{3415}$ kwhr.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: The *mean* calorie is $\frac{1}{100}$ of the heat required to raise the temperature of 1 gram of water from *Zero* C to 100 C. It is substantially equal to the quantity of heat required to raise one gram of water from 14.5 C to 15.5 C.

Central Fan System: A mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and is conveyed to and from the rooms by means of a fan and a system of distribution ducts. See Chapters 9 and 22.

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.*

Column Radiator: A type of direct radiator. This radiator has not been listed by manufacturers since 1926.

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone *(Average):* The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone (Extreme):* The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 2.)

Concealed Radiator: See *Convector.*

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 F between the two surfaces of the material.

Conductor *(heat):* A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also *constant* dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Control: Any manual or automatic device for the regulation of a machine to keep it at normal operation. If automatic, it is considered that the device is motivated by variations in temperature, pressure, time, light, or other influences.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced.*

Convector: A concealed *radiator.* A heating unit and an enclosure or shield located either within, adjacent to, or exterior to the room or space to be heated, but transferring heat to the room or space mainly by the process of convection. If the heating unit is located exterior to the room or space to be heated, the heat is transferred through one or more ducts or pipes; see Chapter 30.

Corrosive: Having the power to wear away or gradually change the texture or substance of a material.

Decibel: The standard unit for noise or sound intensity. One decibel is equal to ten times the logarithm to the base *e* of the ratio of the sound intensities.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exist as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over a 24-hour period, and a temperature of 65 F.

Dehumidify: To remove water vapor from the atmosphere; to remove water vapor or moisture from any material.

Density: The weight of a unit volume, expressed in pounds per cubic foot. ρ (rho) $= \frac{W}{V}$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Diffuser: A vaned device placed at an air supply opening to direct the air flow.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *radiator.*

Direct-Return System *(Hot water):* A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser *(Steam):* A pipe which carries steam downward to the heating units and into which the condensation from the heating units drains.

Down-Feed System *(Steam):* A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head *(Side Outlet Enclosure):* The height of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening.

Draft Head *(Top Outlet Enclosure):* The height of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Dry Air: Air with which no water vapor is mixed. This term is used comparatively, since in nature there is always some water vapor included in air, and such water vapor, being a gas, is dry.

Dry-Bulb Temperature: The temperature of the air indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. See *wet return.*

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: The total or impact pressure. This is the sum of the radial pressure and the velocity pressure at the point of measurement.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index

which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Enthalpy: Total heat or thermal potential.

Entropy: The logarithmic probability of a state. It is the integration between two absolute temperatures of the quotient of the quantity of heat divided by the absolute temperature at the condition at which the temperature is taken. It is, therefore, a numeric which explains a difference in conditions between two points in a heat cycle.

Entropy, which can vary with temperature, volume, or pressure, is constant during adiabatic expansion in a reversible cycle or during isentropic expansion in an irreversible cycle. Entropy is a function of the unavailable energy in any system.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined. It is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Extended Heating Surface: See *Heating Surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

Fan Furnace System: See *Warm Air Heating System.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = \frac{WV}{gt}$.

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a fire-pot.

Furnace Volume *(total):* The total furnace volume for horizontal-return tubular boilers and water-tube boilers is the cubical contents of the

furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal-return tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929.)

Gage Pressure: Pressure measured from atmospheric pressure as a base. Gage pressure may be indicated by a manometer which has one leg connected to the pressure source and the other exposed to atmospheric pressure.

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as defined above. (A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers.)

Gravity Warm Air Heating System: See *Warm Air Heating System.*

Grille: A perforated covering for an air inlet or outlet usually made of wire screen, pressed steel, cast-iron or plaster. Grilles may be plain or ornamental.

Heat: A form of energy generated by the transformation of some other form of energy, as by combustion, chemical action, or friction. According to the molecular theory, heat consists of the kinetic and potential energy of the molecules of a substance. The addition of heat energy to a body increases the temperature or the kinetic energy of motion of its molecules (*sensible heat*) or increases their potential energy of position but does not increase the temperature, as when melting or boiling occurs (*latent heat*).

Heat Capacity: The amount of heat (Btu or calories) required to raise the temperature of a body of any mass and variety of parts one degree (Fahrenheit or centigrade). This will depend on the masses and specific heats of the various parts of the body.

Therefore

$$S = m_1 s_1 + m_2 s_2 + m_3 s_3 \ldots \text{etc.}$$

where

S is the heat capacity and m_1, m_2, m_3, and s_1, s_2, s_3 stand for the masses and corresponding specific heats of the parts, respectively.

Heating Medium: A substance such as water, steam, air, electricity

or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *Boiler Heating Surface.*)

Heat of the Liquid: The sensible heat of a mass of liquid above an arbitrary zero.

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft-lb per second or 33,000 ft-lb per minute. (One horsepower = 745.8 watts. In practice this is considered 746 watts.)

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidity: The water vapor mixed with dry air in the atmosphere. *Absolute humidity* refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. *Specific humidity* refers to the weight of water vapor in pounds carried by one lb of dry air. *Relative humidity* is a ratio, usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. Relative humidity is either the ratio of the actual partial pressure of the water vapor in the air to the saturation pressure at the dry-bulb temperature, or the ratio of the actual density of the vapor to the density of saturated vapor at the dry-bulb temperature. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space.

Humidistat: A regulatory device, actuated by changes in humidity, used for the control of humidity.

Hygrostat: Same as humidistat.

Inch of Water: A measure of pressure which refers to the difference in the heights of the legs of a water filled manometer.

Insulation *(heat):* A material having a relatively high heat-resistance per unit of thickness.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

Latent Heat: See *Heat.*

Laws of Thermodynamics: The *first law* states that the total energy of an isolated system remains constant and cannot be increased or diminished by any physical process whatever. The *second law* states that no change in a system of bodies that takes place of itself can increase the available energy of a system.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light

oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second.

$$m = \frac{W}{g}.$$

Mb, Mbh[1]**:** Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The mechanical energy necessary to produce 1 Btu of heat energy. J = 777.5 ft-lb.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol: The unit of weight for gases. It is defined as m lb where m denotes the molecular weight of a gas. For any gas the volume of 1 *mol* at 32 F and standard atmospheric pressure is 358.65 cu ft and the weight of a cubic foot is 0.002788 m lb.

Neutral Zone: The level within a room or building at which the pressure is exactly equal to the outside barometric pressure.

One-Pipe Supply Riser *(steam):* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System *(hot water):* A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System *(steam):* A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units.

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work, expressed in units of horsepower, one of which is equal to 550 ft-lb of work per second, or 33,000 ft-lb per minute.

These symbols were approved by the A.S.H.V.E., June, 1933.

Prime Surface: See *Heating Surface.*

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects "it can see" and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage. *Concealed Radiator:* See *Convector.*

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Refrigerant: A substance which produces a refrigerating effect by its absorption of heat while expanding or vaporizing.

Register: A grille with a built-in multiblade damper or shutter.

Relative Humidity: See *Humidity:* see also discussion of relative humidity in Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*hot water*)*:* A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major subdivision of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to permit egress of air.

Saturated Air: Air containing as much water vapor as it can hold without any condensing out; in saturated air, the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature.

Sensible Heat: See *Heat.*

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The quantity of heat, expressed in Btu, required to raise the temperature of 1 lb of a substance 1 F.

Specific Volume: The volume, expressed in cu ft, of one pound of a substance. $v = \frac{1}{\rho} = \frac{V}{W}$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*equivalent*): Equivalent direct *radiation* (EDR). By definition, that amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

Stack Height: The height of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: As defined by A.S.H.V.E. codes, *standard air* is air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07495 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.92 in. of mercury. The error involved in disregarding the difference between the above two weights is very slight and in most instances may be neglected.)

Static Pressure: The compressive pressure existing in a fluid. It is a measure of the potential energy of the fluid.

Steam: Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. Steam in contact with the water from which it has been generated may be *dry saturated* steam or *wet saturated* steam. The latter contains more or less actual water in the form of mist. If steam is heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated.*

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *Steam.*

Supply Mains (*steam*): The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection *from a surface to the air or liquid surrounding it*, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Synthetic Air Chart: A chart for evaluating the air conditions maintained in a room.

Thermal Resistance: The reciprocal of *conductance.*

Thermal Resistivity: The reciprocal of *conductivity.*

Thermodynamics: The science which treats of the mechanical actions or relations of heat.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The extraction of 12,000 Btu per hour.

Ton Day of Refrigeration: The heat removed by a ton of refrigeration operating for one day; 288,000 Btu.

Total Heat: A thermodynamic quantity, variously called heat content, thermal potential, enthalpy. It is the heat required per unit mass (Btu per lb) to raise a given substance to a given point from an arbitrary datum point. It is the sum of the heat of the liquid, the latent heat, and any miscellaneous heat which may be present.

Total Pressure: The sum of the static and velocity pressures in a fluid. It is a measure of the total energy of the fluid.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*steam or water*)*:* A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit.

Underfeed Distribution System (*hot water*)*:* A hot water heating system in which the main flow pipe is below the heating units.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit Air Conditioner: A piece of equipment designed to provide simutaneous control of at least four of the seven functions (page 201) involved in summer and winter air conditioning. The apparatus is compactly housed in a cabinet placed within or immediately adjacent to the rooms served. The parts comprising a unit air conditioner are assembled at the point of manufacture, and the performance of the assembly is the responsibility of the manufacturer. See Chapter 12.

Unit Cooler: A cooling device, usually comprising an extended-surface element and a motor-driven fan mounted integrally in a housing, located within or adjacent to the room served. Generally no ducts are attached to inlet or outlet. The refrigerant is brought to the unit from an outside source, and the fan drives air over the cooling element.

Unit Heater: A heating device, usually comprising an extended-surface element or a gas burner, mounted with a motor-driven fan in a housing, located within or adjacent to the room served. Generally, no ducts are attached to inlet or outlet. The fluid for heating is brought to the unit from an outside source, and the fan drives air over the heating element. Unit heaters are used primarily in industrial applications.

Unit Ventilating-Heater: A ventilating and heating device comprising a motor-driven fan, an extended-surface heating element and usually a filter, mounted in a housing, located within or adjacent to the room served. Outdoor air is obtained through a dampered direct connection or a short duct from a nearby wall or window opening. Provision

for partial recirculation is usually made. If a humidifier is included, such a filter-equipped device becomes a winter-type unit conditioner. Unit ventilators are used primarily for offices, schools, and places of public assembly.

Up-Feed System (*steam*): A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: The equilibrium pressure exerted by a vapor in contact with its liquid.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The pressure corresponding to the velocity of flow. It is a measure of the kinetic energy of the fluid.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*.)

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: The lowest temperature which a water wetted body will attain when exposed to an air current. This is the temperature of adiabatic saturation.

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. The wet return usually is below the level of the water line in the boiler, although not necessarily so.

ABBREVIATIONS[2]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Area	*A*
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiler pressure	bp
Boiling point	bp
Brake horsepower	bhp
Brake horsepower-hour	bhp-hr
British thermal unit	Btu
Calorie	cal
Centigram	cg
Centimeter	cm
Centimeter-gram-second (system)	cgs
Change in specific volume during vaporization	v_{fg}
Cubic	cu
Cubic foot	cu ft
Cubic feet per minute	cfm
Cubic feet per second	cfs
Decibel	db
Degree[3]	deg or °
Degree centigrade	C
Degree Fahrenheit	F
Degree Kelvin	K
Degree Réaumur	R
Density, Weight per unit volume, Specific weight	*d* or ρ (rho)

$$\rho = \frac{1}{v}$$

Diameter	*D* or diam
Direct-current (as adjective)	d-c
Distance, linear	*s*
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, Vapor in contact with liquid	*Subscript* g
Entropy (The capital should be used for any weight, and the small letter for unit weight.)	*S* or *s*
Feet per minute	fpm
Feet per second	fps
Foot	ft
Foot-pound	ft-lb
Foot-pound-second (system)	fps
Force, total load	*F*
Freezing point	fp
Gallon	gal
Gallons per minute	gpm
Gallons per second	gps
Gram	g
Gram-calorie	g-cal

[2]From compilations of abbreviations approved by the *American Standards Association*, Z, 10 a, c, f, and i. As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word.

[3]It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Head .... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight.) .... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid .... h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor .... h_g
Heat of vaporization at constant pressure .... L or h_{fg}
Horsepower .... hp
Horsepower-hour .... hp-hr
Inch .... in.
Inch-pound .... in-lb
Indicated horsepower .... ihp
Indicated horsepower-hour .... ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight.) .... U or u
Kilogram .... kg
Kilowatt .... kw
Kilowatthour .... kwhr
Length of path of heat flow, thickness L
Load, total .... W
Mass .... m
Mechanical efficiency .... e_m
Mechanical equivalent of heat .... J
Melting point .... mp
Meter .... m
Micron .... μ (mu)
Miles per hour .... mph
Minute .... min
Molecular weight .... mol. wt
Mol .... mol
Ounce .... oz
Power, Horsepower, Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Revolutions per minute .... rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* f
Specific gravity .... sp gr
Specific heat .... sp ht or c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Square foot .... sq ft
Square inch .... sq in.
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion.) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature.) .... T or Θ (*capital theta*)
Thermal conductance[4] (heat transferred per unit time per degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, Unit conductance (heat transferred per unit time per unit area per degree) .... C_a

$$C_a = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples are—conductivity and resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples are—conductance and transmittance. Terms ending *ion* designate rate of heat transfer. Examples are—conduction and transmission.

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$
Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* fg
Velocity V
Volume (total) V
Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q
Watt .... w
Watthour .... whr
Weight of a major item, Total weight .... W
Weight rate, Weight per unit of power, Weight per unit of time .... w
Work (total) .... W

CONVERSION EQUATIONS

Fahrenheit degrees = 9/5 centigrade degrees + 32.

Centigrade degrees = 5/9 (Fahrenheit degrees − 32).

Absolute temperature, expressed in Fahrenheit degrees = Fahrenheit degrees + 459.6. In heating and ventilating work, 460 is usually used.

Absolute temperature, expressed in centigrade degrees = centigrade degrees + 273.1.

Power, Heat, and Work

1 ton refrigeration	= {12,000 Btu per hour 200 Btu per minute
Latent heat of ice	= 143.33 Btu per pound

1 Btu	= 777.5 ft-lb
	0.293 watthours
	252.02 mean calories
1 watthour	= 2,655.2 ft-lb
	3.415 Btu
	3600 joules
	860.648 mean calories
1 mean calorie	= 0.003968 Btu
	3.085 ft-lb
	0.0011619 watthours
1 kilowatt (1000 watts)	= 1.3405 horsepower
	56.92 Btu per minute
	44,252.7 ft-lb per minute
1 horsepower	= 0.746 kilowatt
	42.44 Btu per minute
	33,000 ft-lb per minute
	550 ft-lb per second
1 boiler horsepower	= 33,471.9 Btu per hour

Weight and Volume

1 gal (U. S.)	= 231 cu in.
	0.13368 cu ft
1 British or Imperial gallon	= 277.274 cu in.
1 cu ft	= 7.4805 gal
	1728 cu in.
1 cu ft water at 60 F	= 62.37 lb
1 cu ft water at 212 F	= 59.76 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.99 lb
1 lb (avdp)	= 16 oz
	7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb
1 long ton	= 2240 lb

Pressure

1 lb per square inch	= 144 lb per square foot
	2.0416 in. mercury at 62 F
	2.309 ft water at 62 F
	27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F
	1.732 in. water at 62 F
1 atmosphere	= 14.7 lb per square inch
	2116.3 lb per square foot
	33.974 ft water at 62 F
	30 in. mercury at 62 F
	29.921 in. mercury at 32 F
1 in. water at 62 F	= 0.03609 lb per square inch
	0.5774 oz per square inch
	5.196 lb per square foot
1 ft water at 62 F	= 0.433 lb per square inch
	62,355 lb per square foot
1 in. mercury at 62 F	= 0.491 lb per square inch
	7.86 oz per square inch
	1.131 ft water at 62 F
	13.57 in. water at 62 F

Metric Units

1 cm	= 0.3937 in.
1 in.	= 2.54 cm
1 m	= 3.281 ft
1 ft	= 0.3048 m
1 sq cm	= 0.155 sq in.
1 sq in.	= 6.45 sq cm
1 sq m	= 10.765 sq ft
1 sq ft	= 0.0929 sq m
1 cu cm	= 0.061 cu in.
1 cu in.	= 16.39 cu cm
1 cu m	= 35.32 cu ft
1 cu ft	= 0.0283 cu m
1 liter	= 1000 cu cm = 0.264 gal
1 kg	= 2.2046 lb
1 lb	= 0.4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 980.59 dynes = 0.002205 lb
1 kilometer per hour	= 0.6214 mph
1 gram per square centimeter	= { 0.0290 in. mercury, at 0 deg C; 0.394 in. water, at 15 C }
1 kg per square centimeter (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= { 0.03614 lb per cubic inch; 62.43 lb per cubic foot }
1 dyne	= 0.00007233 poundals
1 joule	= { 10,000,000 ergs; 0.73767 ft-lb }
1 metric horsepower	= { 75 kg-m per second; 0.986 hp (U. S.) }
1 kilogram-calorie (large calorie)	= { 1000 gram-calories (small calorie); 3.97 Btu }
1 kilogram-calorie per kilogram	= 1.8 Btu per pound
1 gram-calorie per square centimeter	= 3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	= 1.451 Btu per square foot per inch
1 gram-calorie per second per square centimeter for a temperature graduation of 1 deg C per centimeter	= 2903 Btu per hour per square foot for a temperature graduation of 1 deg F per inch of thickness.

SYMBOLS FOR HEATING AND VENTILATING DRAWINGS[5]

1. The objects of this standard set of symbols are to insure the correct interpretation of drawings and to conserve drafting room time by establishing simple and unmistakable symbols for the component parts of the heating and ventilating systems. In preparing the list of symbols an effort has been made to follow existing practice in so far as possible but the list cannot be expected to match exactly the existing practice of every drafting room.

2. Simplicity, ease of execution and unmistakable identification were carefully considered in selecting the symbols. Uncommon fittings and appliances such as vacuum pumps, separators, etc., have purposely been omitted in order to produce a list which can be easily remembered. It is assumed that when the scale of the drawing permits, the valves and fittings will be drawn to scale and a conventional representation is then unnecessary.

[5]From A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, edition of 1929.

3. High pressure steam supply pipe

4. Low pressure steam supply pipe

5. Hot water pipe—flow

6. Return pipe—steam or water

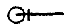
7. Air vent line

8. Flanges 

9. Screwed union 

10. Elbow 

11. Elbow—looking up 

12. Elbow—looking down 

13. Tee 

14. Tee—looking up 

15. Tee—looking down 

16. Gate valve 

17. Globe valve 

18. Angle valve 

19. Angle valve—stem perpendicular 

20. Lock shield valve 

21. Check valve 

22. Reducing valve 

23. Diaphragm valve 

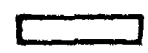
24. Diaphragm valve—stem perpendicular 

25. Thermostat 

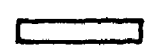
26. Radiator trap—elevation 

27. Radiator trap—plan 

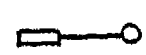
28. Expansion joint 

29. Column radiator—plan 

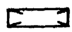
30. Column radiator—elevation 

31. Wall radiator—plan 

32. Wall radiator—elevation 

33. Pipe coil—plan 

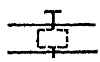
34. Pipe coil—elevation 

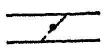
35. Indirect radiator—plan 

36. Indirect radiator—elevation 

37. Supply duct—section 

38. Exhaust duct—section 

39. Butterfly damper—plan (or elevation) 

40. Butterfly damper—elevation (or plan) 

41. Deflecting damper—square pipe 

42. Vanes 

43. Air supply outlet 

44. Exhaust outlet 

A.S.H.V.E. CODES

The following codes and standards relating to the design, installation, testing, rating, and maintenance of materials and equipment used for the heating and ventilation of buildings, have been adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Air Cleaning Devices	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work	June, 1934	A.S.H.V.E. Reprint
Air purity	Synthetic Air Chart	June, 1917	A.S.H.V.E. TRANSACTIONS, Vol. 23, p. 607, and THE GUIDE, 1931
Boilers (testing)	Standard and Short-Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929
Boilers (testing)	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3)[a]	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929
Boilers—Oil Fuel (testing)	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931
Boilers (rating)	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers	January, 1929 Revised April, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42
Concealed Gravity Type Radiation	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section)	June, 1934	A.S.H.V.E. Reprint
Convectors	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)	January, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Ethics	Code of Ethics for Engineers	January, 1922	A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 6 (See frontispiece THE GUIDE, 1935)
Fans	Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers	May, 1923. Revised June, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407[b]
Garages	Code for Heating and Ventilating Garages	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355

[a]Originally adopted by the *National Boiler and Radiator Manufacturers Association.*
[b]Also, see *Heating, Piping and Air Conditioning*, August, 1931, p. 743.

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Heat transmission through walls	Standard Test Code for Heat Transmission through Walls	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253
Minimum requirements	Code of Minimum Requirements for Heating and Ventilation of Buildings, Edition–1929	June, 1925	A.S.H.V.E. Codes
Pitot tube	Code for Use of Pitot Tube	January, 1914	A.S.H.V.E. TRANSACTIONS, Vol. 20, 1914, p. 211
Radiators	Code for Testing Radiators	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18
Unit heaters	Standard Code for Testing and Rating Steam Unit Heaters[c]	January, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25
Vacuum Heating Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps	June, 1934	A.S.H.V.E. Journal; *Heating, Piping and Air Conditioning*, March, 1934, p. 136
Ventilation	Report of Committee on Ventilation Standards	August, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 383

The following Codes and Standards have been endorsed or approved by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	SPONSORED BY	REFERENCE
Chimneys	Standard Ordinance for Chimney Construction	*National Board of Fire Underwriters*	Chapter 14, THE GUIDE, 1931
Piping systems	Identification of Piping Systems[d]	*American Society of Mechanical Engineers*	*Heating, Piping and Air Conditioning*, July, 1929
Warm air furnaces	Standard Code Regulating the Installation of Gravity Warm Air Furnaces in Residences	*National Warm Air Heating Association*	*National Warm Air Heating Association*, Columbus, Ohio

[c]Adopted jointly by the *Industrial Unit Heater Association*, and the A.S.H.V.E.

[d]Adopted November, 1928, Sponsored by (1) *American Society of Mechanical Engineers*, (2) *National Safety Council.*

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3. Industrial Air Conditioning.
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6. Air Leakage.
7. Heating Load.
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10. Cooling Methods.
11. Humidification and Dehumidification.
12. Unit Air Conditioners and Conditioning Systems.
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15. Air Pollution.
16. Air Cleaning Devices.
17. Fans and Motive Power.
18. Sound Control.
19. Air Distribution.
20. Air Duct Design.
21. Industrial Exhaust Systems.
22. Fan Systems of Heating.
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Roll of Membership

AMERICAN SOCIETY *of*
HEATING *and* VENTILATING ENGINEERS

1935

Contains Lists of Members Arranged Alphabetically and Geographically, also Lists of Officers and Committees, Past Officers and Local Chapter Officers

Corrected to January 1, 1935

Published at the Headquarters of the Society
51 Madison Avenue, New York, N. Y.

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A

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ACHESON, Albert R. (*M* 1919), Consulting Engr. (for mail), 852 Ostrom Ave., Syracuse, N. Y.

ADAMS, Benjamin (*M* 1919), Dist. Mgr. (for mail), American Blower Corp., 612 Otis Bldg. and 3006 W. Coulter St., Queen Lane Manor, Philadelphia, Pa.

ADAMS, Charles W. (*M* 1920), Salesman, U. S. Radiator Corp., 1405 West 11th St., Kansas City, Mo.

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BLACK, F. C. (*M* 1919), Pres. (for mail), F. C. Black Co., 622 W. Randolph St., and 4535 N. Ashland Ave., Chicago, Ill.
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BOUCHERLE, Henry N. (*M* 1934), Secy. (for mail), The Scholl-Choffin Co., Mahoning Ave. and Hogue St., and 3412 Hudson Ave., Youngstown, Ohio.
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BRAUN, Louis T. (*M* 1921), Executive Secy. (for mail), Chicago Master Steamfitters Assn., 228 N. LaSalle St., and 1548 Pratt Blvd., Chicago, Ill.
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BRINKER, Harry A. (*M* 1934), Member of Firm, Wilson-Brinker Co., 412 Pythian Bldg., and (for mail), 524 Village St., Kalamazoo, Mich.
BRINTON, Joseph Ward (*M* 1920), Dist. Mgr. (for mail), American Blower Corp., 1003 Statler Bldg., Boston, and 42 Gleason St., West Medford, Mass.
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BROOKS, Frank W. (*S* 1934), (for mail), 2111 Abington Rd., Cleveland, and 935 N. Broadway, Dayton, Ohio.
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BROWN, Morris (*J* 1928), Htg. Engr. (for mail), Brown Bros., 340 Talbot Ave., and 609 Park St., Dorchester, Mass.
BROWN, Ronald F. (*S* 1933), 66 Mitchell Ave., Binghamton, N. Y.

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BROWN, W. Maynard (*A* 1930), Warren Webster & Co., 17th and Federal Sts., Camden, N. J.
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BURKE, James J. (*J* 1930), Engr. (Air Cond.), Carrier Engrg. Corp., 850 Frelinghuysen Ave., Newark, and (for mail), 720 N. Broad St., Elizabeth, N. J.
BURKE, William J. (*A* 1934), 1109 S. Carson St., Tulsa, Okla.
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BUTT, Roderick E. W. (*J* 1930), Partner, Crerar, Butt & Co., 14 Regent St., London S.W.I., and (for mail), 3 Orme Court, London W2, England.
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CAMPBELL, Robert E. (*S* 1934), c/o Mrs. L. Winn, 781 Ocean Ave., Brooklyn, N. Y.
CAMPBELL, Thomas F. (*M* 1928), Minneapolis-Honeywell Regulator Co., 1013 Penn Ave., Wilkinsburg, Pa.
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CAREY, James A. (*M* 1928), Carrier Engrg. Corp., Newark, N. J., and (for mail), Villanova, Pa.
CAREY, Paul C. (*M* 1930), (for mail), Runyon & Carey, 33 Fulton St., Newark, and 31 Claremont Dr., Maplewood, N. J.
CARLE, William E. (*M* 1926), Pres. (for mail), Carle-Boehling Co., Inc., 1641 W. Broad St., and 2220 Floyd Ave., Richmond, Va.
CARLSON, Everett E. (*M* 1932; *A* 1929), Br. Mgr. (for mail), The Powers Regulator Co., 1010 Louderman Bldg., and 6652 Washington Ave., St. Louis, Mo.
CARMAN, George G. (*A* 1931; *J* 1928), Lewis Institute, Chicago, Ill.

CARPENTER, R. H. (*M* 1921), (Council, 1930-1934), Mgr., New York Office (for mail), Nash Engrg. Co., Graybar Bldg., 420 Lexington Ave., New York, and 20 Jefferson Ave., White Plains, N. Y.
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CARRIER, Earl G. (*J* 1929), Estimating Engr., Carrier South Africa (Pty), Ltd., 20 Beresford House, Simmonds St., Johannesburg, Transvaal, Union of South Africa.
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CASEY, Byron L. (*M* 1921), Sales Engr. (for mail), Ilg Electric Vtg. Co., 182 N. LaSalle St., Chicago, and 515 N. Park Ave., Park Ridge, Ill.
CASEY, Huntley F. (*M* 1931), Sales Repr., P. O. Box 271, E. Falls Church, Va., and (for mail), 756 E. River St., Anderson, S. C.
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CHERRY, Lester A.* (*M* 1921), Consulting Engr. (for mail), Industrial Planning Corp., 271 Delaware Ave., Buffalo, and 155 Euclid Ave., Kenmore, Erie Co., N. Y.
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CHRISTENSON, Harry (*A* 1931), Supt. of Htg. (for mail), Hunter Prell Co., 311 Elm St., and 85 Wentworth Ave., Battle Creek, Mich.
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CHRISTMAN, William F. (*A* 1932; *J* 1931), (for mail), Kroeschell Engrg. Co., 2306 N. Knox Ave., and 3912 N. Hoyne Ave., Chicago, Ill.
CHURCH, Herbert John (*M* 1922), Mgr. (for mail), Darling Brothers, Ltd., 137 Wellington St. W., Room 902 Toronto, and 358 Main St. N., Weston, Ont., Canada.
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COX, Harrison F. (*A* 1930), 243 Carroll St., Paterson, N. J.

COX, William W. (*M* 1923), (for mail), Heating Service Co., 326 Columbia St., and 6232–31st Ave. N.E., Seattle, Wash.

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DAVIS, Arthur Folsom (*M* 1934), Vice-Pres. (for mail), The Johnson & Davis Plbg. & Htg. Co., 2235 Arapahoe St., and 1901 Ivanhoe St., Denver, Colo.
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DAVIS, James R. (*S* 1934), 2111 Abington Rd., and (for mail), 9704 Miles Ave., Cleveland, Ohio.
DAVIS, Joseph (*M* 1927; *A* 1926), Owner, Htg. Engr. and Contractor (for mail), 607 Root Bldg., 70 W. Chippewa, and 166 Huntington Ave., Buffalo, N. Y.
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DAVISON, Robert L. (*M* 1934), Director of Research (for mail), John B. Pierce Foundation, 40 West 40th St., and 28 East 10th St., New York, N. Y.
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DEAN, Frank J., Jr. (*S* 1934), Clerical, Automatic Electric Co., 1033 W. Van Buren, and (for mail), 126 S. Central Ave., Chicago, Ill.
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DEWEY, R. P. (*M* 1934), Chief Engr. (for mail), Barber-Colman Co., and 2301 Oxford St., Rockford, Ill.
DIBBLE, S. E.* (*M* 1917), (*Presidential Member*), (Pres., 1925; 1st Vice-Pres., 1924; 2nd Vice-Pres., 1922; Council, 1921-1926), Supt., Thomas Ranken Patton School, Elizabethtown, Pa.
DICE, Eugene S. (*S* 1933), 7141 Upland St., Pittsburgh, Pa.
DICKENSON, Frederick R. (*A* 1934), Dist. Mgr. (for mail), American Blower Corp., 1302 Swetland Bldg., Cleveland, and 3435 Menlo Rd., Shaker Heights, Ohio.
DICKEY, Arthur J. (*M* 1921), 9 Mossom Pl., Toronto, Ont., Canada.
DICKSON, Robert B. (*M* 1919), Pres. (for mail), Kewanee Boiler Co., Inc., Franklin St. and Q Tracks, and 409 E. Prospect St., Kewanee, Ill.
D'IMOR, Elton J. (*M* 1933), Br. Mgr. and Engr., The Trane Co., LaCrosse, Wis., and (for mail), 189 N. Auburndale St., Apt. 7, Memphis, Tenn.
DISNEY, Melvin A. (*A* 1934), Co-Partner, Mfr's. Reprs., Htg., Vtg., and Air Cond. Equip. (for mail), 4301½ Main St., and 8024 Mercier, Kansas City, Mo.
DISTEL, Frank (*M* 1918), Owner, Distel Heating Equipment Co., 404-406 Kalamazoo Plaza (for mail), P. O. Box 133, and 1011 W. Genesee St., Lansing, Mich.
DIVER, M. L. (*M* 1925), Consulting Engr., P. O. Box 1016, San Antonio, Texas.
DIXON, Arthur G. (*M* 1928), Sales Mgr. (for mail), Modine Mfg. Co., and 442 Wolff St., Racine, Wis.
DOBBS, C. E. (*A* 1921), Repr., Burnham Boiler Corp., 31st and Jefferson Sts., Philadelphia, Pa., and (for mail), 72 Berlin Ave., Haddonfield, N. J.
DODDS, Forrest F. (*M* 1920), Br. Mgr. (for mail), American Radiator Co., 1023 Grand Ave., and 235 Ward Pkwy., Kansas City, Mo.
DODGE, Harry G. (*A* 1934), Vice-Pres., Metropolitan Pipe & Supply Co., 145 Broadway, Cambridge, and (for mail), 28 Rustic Rd., Melrose Highland, Mass.
DOERING, Frank L. (*M* 1919), Salesman, American Radiator Co., 219 Denver Ave., Lynchburg, Va.
DOHERTY, Russell (*A* 1929), Chicago Dist. Mgr. (for mail), National Radiator Corp., 1111 East 83rd St., Chicago, and 300 Forest Ave., Oak Park, Ill.
DOLAN, Raymond G. (*M* 1926; *A* 1926; *J* 1922), Secy-Treas. (for mail), Tom Dolan Htg. Co., Inc., 614 W. Grand, and 2112 West 20th, Oklahoma City, Okla.
DONNELLY, James A.* (*M* 1904), (Treasurer, 1912-1914), Largent, W. Va.
DONNELLY, Russell (*M* 1923), Sales Engr. (for mail), Nash Engrg. Co., Graybar Bldg., 420 Lexington Ave., New York, N. Y.
DONOVAN, William J. (*A* 1930), 2239 North 27th St., Philadelphia, Pa.
DONZELLI, Enrico (*M* 1933), Piazza SS Pietro e Lino, No. 4., Milan, Italy.
DORFAN, Morton I. (*M* 1929), Mgr. Dust Collecting Div., Blaw-Knox Co., P. O. Box 1198, and (for mail), 6357 Morrowfield Ave., Pittsburgh, Pa.
DORNHEIM, G. A. (*M* 1912; *J* 1906), 15 Hamilton Ave., Bronxville, N. Y.
DORSEY, Francis C. (*M* 1920), Engr. and Contractor (for mail), Francis C. Dorsey, Inc., 4520 Schenley Rd., Roland Park, and 212 Gittings Ave., Baltimore, Md.
DOSTER, Alexis (*A* 1934), Secy. (for mail), The Torrington Mfg. Co., 70 Franklin St., Torrington, and Litchfield, Conn.
DOUGHTY, Charles John (*M* 1925), Pres. and Managing Director (for mail), C. J. Doughty & Co., Fed. Inc., U. S. A., 30 Brenan Rd., and 1920 Ave. Joffre, Shanghai, China.
DOVOLIS, Nick J. (*S* 1935), 3403 Chicago Ave., Minneapolis, Minn.
DOWNE, Edward R. (*M* 1927), American Gas Products Corp., 40 West 40th St., New York, N. Y.
DOWNE, Henry S. (*Life Member; M* 1895), Cie Nationale des Radiateurs, 149 Blvd. Haussman, Paris, France.
DOWNES, Henry H. (*M* 1923), Mgr. Navy Equip. Div. (for mail), American Blower Corp., 514 Woodward Bldg., Washington, D. C., and 4605 Davidson Dr., Chevy Chase, Md.

DOWNES, Nate W. (*M* 1917), (Council, 1928-1930), Chief Engr. and Supt. of Bldgs. (for mail), School Dist. of Kansas City, 317 Finance Bldg., and 2119 East 68th St., Kansas City, Mo.
DOWNS, Sewell H. (*M* 1931), Chief Engr., Clarage Fan Co., and (for mail), 211 Creston Ave., Kalamazoo, Mich.
DOYLE, William J. (*M* 1920), Factory Mgr., The Williamson Heater Co., 4558 Marburg Ave., and (for mail), 3766 Hyde Park Ave., Cincinnati, Ohio.
DRINKER, Philip* (*M* 1922), Assoc. Prof. (for mail), Harvard School of Public Health, 55 Shattuck St., Boston, and Puddingstone Lane, Newton Center, Mass.
DRISCOLL, William H.* (*M* 1904), (*Presidential Member*), (Pres., 1926; 1st Vice-Pres., 1925; 2nd Vice-Pres., 1924; Treas., 1923; Council, 1918-1927), (for mail), Thompson-Starrett Co., Inc., 444 Madison Ave., New York, N. Y., and 50 Glenwood Ave., Jersey City, N. J.
DuBOIS, Louis J. (*M* 1931), Air Cond. Engr., York Ice Machinery Corp., 117 South 11th St., and (for mail), 7337a Lindell Ave., St. Louis, Mo.
DUBRY, Ernest E. (*M* 1924), Asst. Supt., Central Htg., The Detroit Edison Co., 2000 Second Ave., and (for mail), 9116 Dexter Blvd., Detroit, Mich.
DUDLEY, William Lyle (*M* 1922), Vice-Pres. (for mail), Western Blower Co., 1800 Airport Way, and 814–32nd Ave., Seattle, Wash.
DUFF, Kennedy (*M* 1915), Mgr. (for mail), Johnson Service Co., 28 East 29th St., New York, N. Y., and 9 Park Ave., Maplewood, N. J.
DUGAN, Thomas M. (*M* 1920), Sanitary and Htg. Engr., National Tube Co., Fourth Ave. and Locust St., and (for mail), 1308 Freemont St., McKeesport. Pa.
DUGGER, Earl R. (*S* 1934), 3409 Classen, Oklahoma City, Okla.
DUNCAN, George W., Jr. (*M* 1923), 2512 Benvenue Ave., Berkeley, Calif.
DUNCAN, James R. (*M* 1923), Carrier Australasia, Ltd., 56 Hunter St., Sydney, Australia.
DUNCAN, William A. (*A* 1930), Dist. Service Engr. (for mail), Dominion Oxygen Co., Ltd., 92 Adelaide St. W., and 20 Tyrell Ave., Toronto, Ont., Canada.
DUNHAM, Clayton A.* (*M* 1911), Pres. (for mail), C. A. Dunham Co., 450 E. Ohio St., Chicago, and 150 Maple Hill Rd., Glencoe, Ill.
DURKEE, Merritt E. (*A* 1930), Sales Engr. (for mail), C. A. Dunham Co., 101 Park Ave., New York, and 254 Martine Ave., White Plains, N. Y.
DURNING, Edward H. (*J* 1931), Commercial Sales, Dallas Gas Co., Harwood and Jackson Sts., and (for mail), 1830 Moser St., Dallas, Texas.
DURYEA, Albert A. (*J* 1935; *S* 1933), 151 Belden Point, City Island, N. Y.
DUSOSSOIT, Edmond A. (*M* 1920), Treas. (for mail), Lynch & Woodward, Inc., 320 Dover St., Boston, and 16 Hancock Ave., Newton Center, Mass.
DWYER, Thomas F. (*M* 1923), Mech. Engr. (for mail), Board of Education, 49 Flatbush Ave. Ext., Brooklyn, and 1163 Clay Ave., New York, N. Y.
DYER, Orville K. (*M* 1919), Mgr., Blower Div. (for mail), Buffalo Forge Co., 490 Broadway, and 11 Russell Ave., Buffalo, N. Y.

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EAGAR, R. Frank (*M* 1922), 93 Edward St., Halifax, N.S., Canada.
EAKINS, Walter (*M* 1928), 336 E. Phil Ellena St., Germantown, Philadelphia, Pa.
EASTMAN, Carl B. (*M* 1932; *A* 1932; *J* 1929), Mgr. Philadelphia Sales Office, C. A. Dunham Co., 1500 Walnut St., Philadelphia, and (for mail), 7247 Calvin Rd., Upper Darby, Pa.
EASTWOOD, E. O. (*M* 1921), (Council, 1931-1934), Prof. of Mech. Engrg. (for mail), University of Washington, and 4702–12th Ave. N.E., Seattle, Wash.
EATON, Byron K. (*M* 1920), 75 N. Park Rd., La Grange, Ill.
EATON, William G. M. (*A* 1934), Sales Engr., Pease Foundry Co., Ltd., 118 King St. E., Toronto 2, and (for mail), 59 Symington Ave., Toronto 9, Canada.
EBERT, William A. (*M* 1920), Mech. Contractor (for mail), 1026 W. Ashby, and 2151 W. Kings Highway, San Antonio, Texas.
EDWARDS, Daniel F. (*M* 1920), 2340-42 Pine St., St. Louis, Mo.
EDWARDS, Don J. (*A* 1933), Vice-Pres. (for mail), General Heat & Appliance Co., 94 Massachusetts Ave., Boston, and 40 Rockledge Rd., Newton, Mass.
EDWARDS, Paul A. (*M* 1919), Pres. (for mail), The G. F. Higgins Co., 608 Wabash Bldg., and 3074 Pinehurst Ave., Pittsburgh, Pa.
EELLS, Henry B. (*M* 1926), New York Mgr., Barnes & Jones, Inc., 101 Park Ave., New York, and (for mail), 1049 East 27th St., Brooklyn, N. Y.
EGGLESTON, Lewis W. (*M* 1921), American Radiator Co., 5961 Lincoln Ave., Detroit, Mich.
EGGLY, Harry J., Jr. (*M* 1933), Consulting Engr. (for mail), 1805 Walnut St., Philadelphia, and Elkins Park Apts., Elkins Park, Pa.
EHRLICH, M. William* (*M* 1916), Chief Engr., Commodore Heaters Corp., 11 West 42nd St., New York, N. Y., and (for mail), 56 Ridge Rd., Lyndhurst, N. J.
EICHBERG, W. Roy (*M* 1929), Pres. (for mail), Carolina Sheet Metal Corp., 4210 Sansom St., Philadelphia, and 828 Turner Ave., Drexel Hill, Pa.
EICHER, HuBert C. (*M* 1922), State Director, School Bldgs., Div. Dept. of Public Instruction, State Capitol, and (for mail), 103 South St., Harrisburg, Pa.
EISS, Robert M. (*M* 1933; *A* 1933; *J* 1930), (for mail), 72 Brantwood Rd., c/o Buffalo N. Y. P. O., Eggertsville, N. Y.
ELLINGWOOD, Elliott L. (*M* 1909), 354 S. Spring St., Los Angeles, Calif.
ELLIOT, Edwin (*M* 1929), (for mail), Edwin Elliot & Co., 560 North 16th St., Philadelphia, and 403 W. Price St., Germantown, Philadelphia, Pa.
ELLIOTT, Louis (*M* 1932), Consulting Mech. Engr., Electric Bond & Share Co., 2 Rector St., Room 1914, New York, N. Y.
ELLIOTT, Norton B. (*M* 1934), Br. Mgr., American Blower Corp., 1011 Majestic Bldg., and (for mail), 5170 N. Idlewild Ave., Milwaukee, Wis.
ELLIS, Ernest E. (*M* 1922), Secy-Treas., F. A. Ellis & Co., Inc., 840 Center St., Winnetka, Ill.
ELLIS, Frederick E. (*M* 1923), Sales Mgr. (for mail), Imperial Iron Corp., Ltd., 30 Jefferson Ave., Toronto, and 9 Princeton Rd., Kingsway P. O. Toronto 3, Ontario, Canada.
ELLIS, Frederick R. (*M* 1913), Sales Engr., Buerkel & Co., Inc., 18-24 Union Park St., Boston, and (for mail), 131 Beacon St., Hyde Park, Mass.
ELLIS, Harry W. (*M* 1923; *A* 1909), Pres., Johnson Service Co., 507 E. Michigan St., Milwaukee, Wis.
EMERSON, Ralph R. (*M* 1922), 48 Gay St., Newtonville, Mass.
EMERY, Hugh (*J* 1935; *S* 1933), 20 Morgan Pl., N. Arlington, N. J.
EMMERT, Luther D. (*M* 1919), Repr. (for mail), Buffalo Forge Co., Room 1909, 20 N. Wacker Dr., Chicago, and 1704 Hinman Ave., Evanston, Ill.
EMMONS, Neal L. (*S* 1934), 1100 East 19th, Oklahoma City, Okla.
ENGEL, Edward (*J* 1933), Design Draftsman (for mail), Hull Div., U. S. Navy Yard, and 2608 North 30th St., Philadelphia, Pa.

ENGLE, Alfred (*A* 1923), Sales Mgr. (for mail), Jenkins Bros., 80 White St., New York, and 1 Edgewood Rd., Scarsdale, N. Y.
EPPLE, Arnet B. (*J* 1934), Student Engr., B. F. Sturtevant Co., Boston, and (for mail), 8 Elm St., Hyde Park, Mass.
EPPRIGHT, John O. (*J* 1934), 3928 Benton Blvd., Kansas City, Mo.
ERDLE, Gardner F. (*A* 1933), Mfrs. Repr. (for mail), 374 Delaware Ave., Buffalo, and 19 Tremont Ave., Kenmore, N. Y.
ERICKSON, Harry H. (*A* 1929), Sales Engr., General Fittings Co., 804 Architects Bldg., and (for mail), 5909 North 21st St., Philadelphia, Pa.
ERICKSON, Martin E. (*A* 1926), Supt., Maintenance, Board of Education, and (for mail), 1533 South 74th St., West Allis, Wis.
ERICSSON, Eric B. (*M* 1933), Engr., Custodian, Board of Education, and (for mail), 605 West 116th St., Chicago, Ill.
EVANS, C. A. (*M* 1919), 527 Massachusetts Ave., Buffalo, N. Y.
EVANS, Edwin C. (*M* 1919), Consulting Engr., 2953 Zephyr Ave., Corliss Station P. O., Pittsburgh, Pa.
EVANS, William A. (*M* 1918), Mfr. and Agent, W. A. Evans & Co., 180 Broadway, New York, N. Y., and (for mail), 24 Woodland Rd., Maplewood, N. J.
EVELETH, Charles F.* (*M* 1911), 2030 East 115th St., Cleveland, Ohio.
EVERETTS, John, Jr.* (*A* 1935; *J* 1929), Engr. (for mail), W. L. Fleisher, 11 West 42nd St., New York, and 83 Kenilworth Pl., Brooklyn, N. Y.
EVLETH, Everett B. (*A* 1927), Div. Mgr. (for mail), Minneapolis-Honeywell Regulator Co., 43 E. Ohio, Chicago, and 1023 Ashland Ave., Wilmette, Ill.

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FAGIN, Daniel J. (*M* 1932), Heating Engr., House Htg. Div., Laclede Gas Light Co., 11th and Olive, and (for mail), 4920 Chippewa Ave., St. Louis, Mo.
FAHNESTOCK, Maurice K.* (*M* 1927), Research Asst. Prof. (for mail), University of Illinois, 214 M. E. Laboratory, and 701 W. California St., Urbana, Ill.
FAILE, Edward H. (*M* 1934), Consulting Engr., E. H. Faile Co., 441 Lexington Ave., New York, N. Y., and (for mail), 1336 Fairfield Ave., Bridgeport, Conn.
FALTENBACHER, Harry J. (*M* 1930), 235 E. Wister St., Philadelphia, Pa.
FALVEY, John D. (*M* 1922), 6636 Pershing Ave., University City, Mo.
FAMILETTI, A. Robert (*J* 1930), 2812 Wharton St., Philadelphia, Pa.
FANSLER, P. E. (*A* 1927), Editor (for mail), Oil Heat, 167 Madison Ave., New York, N. Y., and Catonsville, Md.
FARLEY, W. F. (*M* 1930), Salesman, American Radiator Co., 40 West 40th St., New York, and (for mail), 28 Elm St., New Rochelle, N. Y.
FARLEY, Willoughby S. (*J* 1933), Partner, Farley & Luther, 120 S. Union St., and (for mail), 211 Montague St., Danville, Va.
FARNHAM, Roswell (*M* 1920), (Council, 1927-1932), Dist. Mgr., Engrg. Sales (for mail), Buffalo Forge Co., P. O. Box 985, and 5 Clarendon Pl., Buffalo, N. Y.
FARNSWORTH, John G. (*J* 1931), Gas House Htg. Engr. (for mail), Central Illinois Light Co., 316 S. Jefferson St., and 313 Crescent St., Peoria, Ill.
FARRAR, Cecil W. (*M* 1920; *A* 1918), (Treas., 1930; Council, 1930), Pres. (for mail), Excelso Products Corp., 1807 Elmwood Ave., and 29 Oakland Pl., Buffalo, N. Y.
FAUST, Frank H.* (*J* 1930), Engr., Air Cond. Dept. (for mail), General Electric Co., 1 River Rd., and 114 Union St., Schnectady, N. Y.
FAY, Francis C. (*M* 1925), Raisler Heating Co., 129-31 Amsterdam Ave., New York, N. Y.
FEBREY, Ernest J. (*M* 1903), Htg. and Air Cond. (for mail), 616 New York Ave. N.W., and 2331 Cathedral Ave., Washington, D. C.
FEEHAN, John B. (*M* 1923), Pres. and Treas. (for mail), John B. Feehan, Inc., 471 Union St., Lynn, and 4 Ocean View Dr., Marblehead, Mass.
FEELY, Frank J. (*A* 1929), 17215 Greenlawn, Detroit, Mich.
FEGLEY, Donald R. (*J* 1935; *S* 1933), 2274 Loring Pl., New York, N. Y.
FEHLIG, John B. (*M* 1918), Pres. (for mail), Excelsior Htg. Supply Co., 528 Delaware St., and 2927 Brooklyn Ave., Kansas City, Mo.
FELDMAN, A. M.* (*Life Member; M* 1903), Consulting Engr., 40 West 77th St., New York, N. Y.
FELS, Arthur B. (*M* 1919), The Fels Co., 42 Union St., Portland, Maine.
FELTWELL, Robert H. (*M* 1905), Htg. Engr., U. S. Radiator Corp., 2321 Fourth St. N.E., and (for mail), 1370 Oak St. N.W., Washington, D. C.
FENNER, N. Paul (*A* 1928), Hoffman Specialty Co., 500 Fifth Ave., Room 3324, New York, N. Y.
FENSTERMAKER, Sidney E. (*M* 1909), Pres. (for mail), S. E. Fenstermaker & Co., 937 Architects and Builders Bldg., and 3102 Washington Blvd., Indianapolis, Ind.
FERGUSON, Ralph R. (*M* 1934; *A* 1927; *J* 1925), Mgr. Trade Dept., American Blower Corp., 401 Broadway, New York, N. Y., and (for mail), 160 Prospect St., East Orange, N. J.
FERNALD, Henry B., Jr. (*J* 1935; *S* 1933), 145 Lorraine Ave., Upper Montclair, N. J.
FERRERO, Henry J. (*J* 1935; *S* 1933), 1738 Adams St., New York, N. Y.
FIEDLER, Harry William (*M* 1923), Pres. (for mail), Air Conditioning Utilities, Inc., 489 Fifth Ave., New York, and 49 Palmer Ave., Scarsdale, N. Y.
FIFE, George D. (*A* 1931; *J* 1929), Engr., Air Cond., National Broadcasting Co., 30 Rockefeller Plaza, and (for mail), 102 East 22nd St., New York, N. Y.
FILKINS, Harry L. (*A* 1932), Vice-Pres., City Ice Co. of Kansas City, 21st and Campbell Sts., and (for mail), 34 East 55 Terrace, Kansas City, Mo.
FILLO, Frank B. (*A* 1934), Minneapolis-Honeywell Regulator Co., 2831 Olive St., St. Louis, Mo.
FINAN, James J. (*M* 1923), Supervising Engr., Board of Education, City of Chicago, 228 N. LaSalle St., Builders Bldg., and (for mail), 7149 Euclid Ave., Chicago, Ill.
FINCH, Stanley B. (*A* 1931), Industrial Engr., Brooklyn Union Gas Co., 180 Remsen St., Brooklyn, N. Y.
FIRESTONE, James F. (*A* 1925; *J* 1914), Exec. Vice-Pres., Round Oak Furnace Co., and (for mail), 203 Orchard St., Dowagiac, Mich.
FITTS, Charles D. (*M* 1920), Mgr. (for mail), American Radiator Co., 692 Prior Ave., St. Paul, and 2807 Dean Blvd., Minneapolis, Minn.
FITTS, Joseph C. (*M* 1930), Secy., Heating, Piping and Air Conditioning Contractors National Assn., 1250 Sixth Ave., New York, N. Y., and (for mail), 215 Kenilworth Rd., Ridgewood, N. J.
FITZGERALD, Matthew J. (*M* 1934), Treas., Standard Asbestos Mfg. Co., 820 W. Lake St., and (for mail), 7314 Harvard Ave., Chicago, Ill.
FITZSIMONS, J. Patrick (*J* 1934; *S* 1932), The Robert Fitzsimons Co., Ltd., 21 Rebecca St., Hamilton, Ont., Canada.
FLANAGAN, Edward T. (*A* 1929), C. A. Dunham Co., Ltd., 1139 Bay St., Toronto, Ont., Canada.
FLARSHEIM, Clarence A. (*J* 1933), Mgr., Air Cond. Dept., Stewart Warner-Alemite Co., 2425 McGee St., and (for mail), 3720 Holmes St., Kansas City, Mo.
FLEISHER, Walter L.* (*M* 1914), Consulting Engr. (for mail), 11 West 42nd St., New York, and Saw Mill Farm, New City, N. Y.

FLEMING, James P. (*M* 1923), Engr.-Custodian, Board of Education, 5045 N. Kimball Ave., Chicago, Ill.
FLINK, Carl H. (*M* 1923), Director of Research (for mail), American Gas Products Corp., 408 East 111th St., New York, and 74 Brookside Ave., Mt. Vernon, N. Y.
FLINT, Coll T. (*M* 1919), N.E. Sales Mgr. (for mail), The H. B. Smith Co., 640 Main St., Cambridge, and 56 Brantwood Rd., Arlington, Mass.
FLOYD, Morris (*M* 1933), Mgr., Air Cond. Div., Edwards Mfg. Co., Cincinnati, Ohio.
FOGARTY, Orville A. (*M* 1934), Mgr., Oil Burner Div., Canadian Fairbanks-Morse Co., Ltd., 980 St. Antonie St., and (for mail), 2178 Old Orchard Ave., Montreal, Que., Canada.
FONDA, Bayard P. (*M* 1934), Air Cond. Engr. (for mail), Bryant Heater Co., 17825 St. Clair Ave., Cleveland, and 2905 Hampton Rd., Shaker Heights, Ohio.
FORFAR, Donald M. (*M* 1917), Mech. Engr. (for mail), Grinnell Co., 240 Seventh Ave. S., and 4817 Emerson Ave. S., Minneapolis, Minn.
FORSBERG, William (*M* 1919), Hopson & Chapin Mfg. Co., 231 State St., New London, Conn.
FORSYTH, Arthur T. (*A* 1934), Dist. Repr., Buffalo Forge Co., 2434 First Ave. S., Seattle, Wash.
FOSTER, Charles (*M* 1923), Consulting Engr. (for mail), 508 Sellwood Bldg., and 2831 E. First St., Duluth, Minn.
FOSTER, James M. (*M* 1930; *A* 1920), Factory Repr. (for mail), 4526 Olive St., St. Louis, and 7021 Lindell Ave., University City, Mo.
FOSTER, Tillman R. (*J* 1930), Carrier Engrg. Corp., 180 N. Michigan Ave., Chicago, Ill.
FOULDS, P. A. L. (*M* 1916), Mech. Engr. (for mail), Office of Hollis French, Consulting Engr., 210 South St., Boston, and 72 Whitin Ave., Point of Pines, Revere, Mass.
FOULDS, Samuel T. N. (*J* 1930), Sales Engr., Power Equipment Co., 791 Tremont St., Boston, and (for mail), 72 Whitin Ave., Revere, Mass.
FOWLES, Harry H. (*J* 1934), Heating Engr., Carman-Thompson Co., 12-14 Lincoln St., Lewiston, and (for mail), Y. M. C. A., Auburn, Maine.
FOX, Otto (*M* 1931), Chief Engr. (for mail), Bryant Heater Co., 17825 St. Clair Ave., Cleveland, and 1819 Fannington Rd., East Cleveland, Ohio.
FRAMPTON, Alfred C. (*S* 1934), 729½ Wilson, Norman, Okla.
FRANK, John M. (*M* 1918; *A* 1912), Ilg Elec. Vtg. Co., 2850 N. Crawford Ave., Chicago, Ill.
FRANK, Olive E.* (*M* 1919), Pres. (for mail), Frank Engrg. Co., 11 Park Pl., and 609 West 114th St., New York, N. Y.
FRANKEL, Gilbert S. (*M* 1926), Mgr., Federal & Marine Dept. (for mail), Buffalo Forge Co., 403 Commercial National Bank Bldg., and 2749 Macomb St. N.W., Washington, D. C.
FRANKLIN, Ralph S. (*M* 1919), Pres-Treas. (for mail), Albert B. Franklin, Inc., 38 Chauncy St., Boston, and 320 Grove St., Melrose, Mass.
FREAS, Royal Bruce (*M* 1928), Pres. (for mail), Freas Thermo Electric Co., 1206 S. Grove St., Irvington, N. J., and 4 West 43rd St., New York, N. Y.
FREEMAN, Alton M. (*A* 1929), Sales Engr., 6088 Plankington Bldg., and (for mail), 4533 N. Bartlett Ave., Milwaukee, Wis.
FREITAG, Frederic G. (*M* 1932), Consulting Engr. (for mail), Sylvestre Oil Co., 709 S. Columbus Ave., and 9 Harrison St., Mt. Vernon, N. Y.
FRENCH, Donald (*M* 1926), Vice-Pres. (for mail), Carrier Corp., 850 Frelinghuysen Ave., Newark, and 45 Waldron Ave., Summit, N. J.
FREY, George O. (*J* 1934), Elec. Engr. (for mail), Warner Bros. Theatres, Inc., 321 West 44th St., New York, and 221 Linden Blvd., Brooklyn, N. Y.
FRIEDMAN, Ferdinand J. (*M* 1921), McDougall & Friedman, 31 Union Square, New York, N. Y., and (for mail), 1221 Osborne St., Montreal, Que., Canada.
FRIEDMAN, Milton (*S* 1933), 470 West End Ave., New York, N. Y.
FRITZ, Charles V. (*S* 1933), P. O. Box 303, Carnegie Institute of Technology, Pittsburgh, Pa.
FRITZBERG, L. Hilding (*J* 1931), Engr. (for mail), B. F. Sturtevant Co., and 1338 River St., Hyde Park, Boston, Mass.
FUKUI, Kunitaro (*M* 1926), Oriental Carrier Engrg. Co., Ltd., Osaka Mitsui Bldg., Nakanoshima, Osaka, Japan.

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GALLIGAN, Andrew B. (*M* 1921), 716 South 51st St., Philadelphia, Pa.
GALLOWAY, James F. (*S* 1934), 176 Clarkson Ave., Brooklyn, N. Y.
GAMMILL, Oscar E., Jr. (*J* 1930), Sales Engr. (for mail), Carrier Engrg. Corp., 1416 Hibernia Bank Bldg., and 2133 Calhoun St., New Orleans, La.
GANT, H. P.* (*M* 1915), (*Presidential Member*), (Pres., 1923; 1st Vice-Pres., 1922; 2nd Vice-Pres., 1921; Council, 1918-1924), Vice-Pres. (for mail), Carrier Engrg. Corp., 12 South 12th St., and Penn Athletic Club, Philadelphia, Pa.
GARDNER, S. Franklin (*M* 1911), Pres. (for mail), Standard Engrg. Co., 2129 Eye St. N.W., and 4901 Hillbrook Lane, Washington, D. C.
GARDNER, William, Jr. (*A* 1921), Vice-Pres. (for mail), Garden City Fan Co., 1842 McCormick Bldg., and 7836 Loomis Blvd., Chicago, Ill.
GARNEAU, Léo (*J* 1930), Sales Engr., Room 743 Dominion Square Bldg., and (for mail), 8454 Brouages St., Montreal, P. Q., Canada.
GAULT, George W. (*S* 1934), Marysville, Pa.
GAUSMAN, Carl E. (*M* 1923), Mech. Engr., 1100 Minnesota Bldg., and (for mail), 2360 Chilcombe Ave., St. Paul, Minn.
GAUTESEN, Alf (*J* 1935; *S* 1933), 1039–79th St., Brooklyn, N. Y.
GAWTHROP, Fred H. (*M* 1919), Pres., Gawthrop & Bro. Co., 705 Orange St., and (for mail), 2211 Shallcross Ave., Wilmington, Del.
GAY, Lewis M. (*A* 1934), Power Engr. (for mail), Texas Power & Light Co., Box 962, Dallas, and 724 Griffith Ave., Terrell, Texas.
GAYLOR, William S. (*M* 1919), Consulting Engr., Flameking Co., Inc., 2159 Madison Ave., New York, and (for mail), 42 Mayhew Ave., Larchmont, N. Y.
GAYLORD, F. H. (*M* 1921), Western Sales Mgr. (for mail), Hoffman Specialty Co., Inc., 130 N. Wells St., Chicago, and 362 N. York St., Elmhurst, Ill.
GEIGER, Irvin H. (*M* 1919), Reg. Prof. Engr. and Mfrs. Repr., Room 319 Telegraph Bldg., Harrisburg, Pa.
GEISSBUHLER, John O. (*S* 1934), University Circle, and (for mail), 9820 Zimmer Ave., Cleveland, Ohio.
GELB, Amiel (*S* 1935), 1042 Irving Ave. N., Minneapolis, Minn.
GENCHI, Bernard (*J* 1935; *S* 1933), 8808–15th Ave., Brooklyn, N. Y.
GERMAIN, Oscar (*M* 1935), Germain Frere, Ltd., 1343 Blvd. St. Louis, Three Rivers, Que., Canada.
GERRISH, Grenville B. (*J* 1930), Mgr., Fitzgibbons Boiler Co., Inc., 80 Boylston St., Boston, and (for mail), 1 Overlook Rd., Melrose, Mass.
GERRISH, Harry E. (*M* 1910), (Council, 1919), Vice-Pres. (for mail), Morgan-Gerrish Co., 307 Essex Bldg., and 4534 Fremont St., Minneapolis, Minn.

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GETSCHOW, George M. (*M* 1906), Pres. and Treas. (for mail), Phillips Getschow Co., 32 W. Austin Ave., and 4542 Beacon St., Chicago, Ill.
GETSCHOW, Roy M. (*M* 1919), Secy. (for mail), Phillips Getschow Co., 32 W. Austin Ave., and 1336 Arthur Ave., Chicago, Ill.
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GIBBS, Frank C. (*M* 1921), Gen. Supt. (for mail), National Regulator Co., 2301 Knox Ave., Chicago, and 150 N. Cuyler Ave., Oak Park, Ill.
GIESECKE, Frederick E.* (*M* 1913), (Council, 1932-1934), Director, Texas Engrg. Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
GIFFORD, Clarence A. (*A* 1934), Salesman, American Radiator Co., 374 Delaware Ave., and (for mail), 247 North Dr., Buffalo, N. Y.
GIFFORD, Robert L. (*M* 1908), Pres., Illinois Engrg. Co., 21st St. and Racine Ave., Chicago, Ill., and (for mail), 1231 S. El Molino Ave., Pasadena, Calif.
GIGUERE, George H. (*M* 1920), Mech. Engr., 800 Marquette Bldg., and (for mail), 17205 Fairport Ave., Detroit, Mich.
GILES, Alfred F. (*J* 1934), H. H. Robertson Co., 2000 Grant Bldg., and (for mail), 4307 Ludwick St., Pittsburgh, Pa.
GILFRIN, George F. (*M* 1932), Gen. Repr. (for mail), Carrier-Brunswick-International, Inc., Apartado 63, Bis and Paseo de la Reforma 120, Lomas de Chapultepec, Mexico, D. F.
GILL, John W. (*S* 1935), 2120 Carter Ave., St. Paul, Minn.
GILLE, Hadar B. (*M* 1930), Skoldungagatan 4, Stockholm, Sweden.
GILLETT, M. C. (*M* 1916), Engr., 6600 Rising Sun Ave., Philadelphia, Pa.
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GILLING, William F., Jr. (*Life Member; M* 1933; *A* 1919), Asst. Mgr., American Radiator Co., 127 Federal St., Boston, and (for mail), 29 Abbott Rd., Wellesley Hills, Mass.
GILMAN, Franklin W. (*M* 1935), Plant Engr. (for mail), Atwater Kent Mfg. Co., 4700 Wissahickon Ave., and 514 W. Coulter St., Philadelphia, Pa.
GILMORE, Louis A. (*J* 1935; *S* 1930), Vice-Pres. (for mail), John Gilmore & Co., 13 North 10th St., and 6186 Westminster Pl., St. Louis, Mo.
GILMOUR, Alan B. (*A* 1932), Salesman, B. F. Gilmour Co., Inc., 152–41st St., and (for mail), 416 Ocean Ave., Brooklyn, N. Y.
GINI, Aldo (*M* 1933), via Correggio 18, Milano, Italy.
GIVIN, Albert W. (*A* 1925), The Gurney Foundry Co., Ltd., P. O. Box 1149, Montreal, P. Q., Canada.
GLANZ, Edward (*A* 1930), Pres. (for mail), Glanz & Killian Co., 1761 W. Forrest Ave., and 3865 Lakewood Ave., Detroit, Mich.
GLASS, William (*M* 1934), Mgr. (for mail), Partridge-Halliday, Ltd., 144 Lombard St., Winnipeg, and 190 Braemar Ave., Norwood, Manitoba, Canada.
GLASSEY, J. Wilbur (*M* 1922), Partner (for mail), Vapor Engrg. Co., 10 South 18th St., Philadelphia, and 7818 Ardleigh St., Chestnut Hill, Philadelphia, Pa.
GLEASON, Gilbert H. (*M* 1923), Partner (for mail), Gilbert Howe Gleason & Co., 25 Huntington Ave., Boston, and 10 Edgehill Rd., Winchester, Mass.
GLORE, Evins Foree (*A* 1916), Pres., Evins F. Glore Sales Corp., 1949 Grand Central Terminal, and (for mail), 644 Riverside Dr., New York, N. Y.
GOELZ, Arnold H. (*M* 1931), Pres. (for mail), Kroeschell Engrg. Co., 2306 N. Knox Ave., Chicago, and 827 Greenwood Ave., Wilmette, Ill.
GOENAGA, Roger C. (*M* 1931), Tech. Director (for mail), Ateliers Ventil., 109 Cours Gambetta, Lyon, and 33 Avenue Valioud, Ste Foy-les-lyon, Rhone, France.
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GOMBERS, Henry B. (*Life Member; A* 1901), Secy. Emeritus, Heating, Piping and Air Conditioning Contractors National Assn., 1250 Sixth Ave., New York City Assn., 1945 Grand Central Terminal Bldg., New York, N. Y., and (for mail), 160 Halsted St., East Orange N. J.
GOODRICH, Charles F. (*M* 1919), Andrews & Goodrich, Inc., Boston, and (for mail), 336 Adams St., Dorchester, Mass.
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GOODWIN, Walter C. (*M* 1933), Div. Engr., Air Cond. Equip. Div. (for mail), Supply Engrg. Dept., Westinghouse Elec. & Mfg. Co., East Pittsburgh, and 6032 Marie St., Pittsburgh, Pa.
GORDON, Edward B., Jr. (*M* 1908), Pres., Pillsbury Engrg. Co., 1200 Second Ave., and (for mail), 2450 West 24th St., Minneapolis, Minn.
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GOTTWALD, C. (*A* 1916), Pres. (for mail), The Ric-wiL Co., Union Trust Bldg., Cleveland, and 2225 Stillman Rd., Cleveland Heights, Ohio.
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GRAHAM, William D. (*M* 1929; *A* 1925; *J* 1923), Dist. Mgr. (for mail), Carrier Engrg. Corp., 800 Union Trust Bldg., Cleveland, Ohio.
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GRANSTON, Ray O. (*J* 1935; *S* 1930), Engr., Univ. Plbg. & Htg. Co., 3939 University Way, and (for mail), 4558 Fourth Ave. N.E., Seattle, Wash.
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GRAVES, Willard B. (*Life Member; M* 1906), Pres. (for mail), W. B. Graves Htg. Co., 162 N. Desplaines St., Chicago, Ill.
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GRAY, William E. (*M* 1922), Sales Engr., Powers Regulator Co., 2720 Greenview Ave., Chicago, Ill., and (for mail), Box 264, High Point, N. C.

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GREEN, **William C.** (*Life Member; M* 1906), Dist. Mgr. (for mail), Warren Webster & Co., 704 Race St., and 244 Erkenbrecher Ave. (Avondale), Cincinnati, Ohio.
GREENBURG, **Dr. Leonard**[s] (*M* 1932), Acting Health Officer (for mail), New Haven Dept. of Health, City Hall, 161 Church St., and 519 George St., New Haven, Conn.
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GRIFFIN, **John J.** (*M* 1921; *A* 1918), Vice-Pres. and Dir. (for mail), Mutual Bank and Trust Co., 716 Locust St., and 3852 Castleman Ave., St. Louis, Mo.
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GROSSMANN, **Harry A.** (*M* 1931), H. A. Grossmann Co., 3221 Olive St., and (for mail), 3122 Geyer Ave., St. Louis, Mo.
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GURNEY, **Edward Holt** (*M* 1929), (Council, 1931-1934), Pres. (for mail), Gurney Foundry Co., Ltd., 4 Junction Rd., and 347 Walmer Rd., Toronto, Ont., Canada.

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HAATVEDT, **Sheldon R.** (*S* 1935), 315-19th Ave. S.E., Minneapolis, Minn.
HACKETT, **H. Berkeley** (*M* 1921), 901 Architects Bldg., 17th and Sansom Sts., Philadelphia, Pa.
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HADEN, **William Nelson** (*Life Member; M* 1902), Late Chairman, G. N. Haden & Sons, Ltd., St. Georges Works, and (for mail), Arnolds Hill, Trowbridge, Wilt, England.
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HAIGNEY, **John E.** (*J* 1935; *S* 1933), 8621 Shore Rd., Brooklyn, N. Y.
HAINES, **John J.** (*M* 1915), Pres. (for mail), The Haines Co., 1933 W. Lake St., Chicago, and 623-17th Ave., Maywood, Ill.
HAJEK, **William J.** (*M* 1932), Br. Mgr. (for mail), Minneapolis-Honeywell Regulator Co., 285 Columbus Ave., and 333 Beacon St., Boston, Mass.
HAKES, **Leon M.** (*M* 1932; *A* 1932; *J* 1929), Sales Engr. (for mail), The R. T. Coe Co., 400 Reynolds Arcade Bldg., and 71 Stratmore Dr.-Greece, Rochester, N. Y.
HALE, **John F.** (*M* 1902), (*Presidential Member*), (Pres., 1913; 1st Vice-Pres., 1912; Board of Governors, 1908-1910, 1912-1913), Dist. Mgr. (for mail), Aerofin Corp., 111 W. Washington St., Rm. 1058, Chicago, and 408 S. Brainard Ave., LaGrange, Ill.
HALEY, **Harry S.*** (*M* 1914), Consulting Engr., Partner (for mail), Leland & Haley, 58 Sutter St., and 735-21st Ave., San Francisco, Calif.
HALL, **John R.** (*J* 1932), Mech. Engr., U. S. Air Cond. Corp., 2101 N.E. Kennedy St., and (for mail), 1416 Lakeview Ave., Minneapolis, Minn.
HALL, **Mora S.** (*M* 1934), Combustion Engr. (for mail), May Oil Burner Corp., Maryland and Oliver St., Baltimore, and Route No. 3, Westminster, Md.
HAMBURGER, **Fred G.** (*J* 1935; *S* 1933), 185 West 102nd St., New York, N. Y.
HAMENT, **Louis** (*A* 1933), Mgr. (for mail), Aquatic Chemical & Metallurgical Engrs., 118 East 28th St., and 568 East 166th St., New York, N.Y.
HAMERSKI, **Francis D.** (*J* 1934), 626 E. Fifth St., Winona, Minn.
HAMILTON, **James E.** (*A* 1933), Mgr. (for mail), U. S. Radiator Corp., 4004 Duncan Ave., St. Louis, and 7715 Shirley Dr., Clayton, Mo.
HAMLIN, **Chauncey J., Jr.** (*A* 1934), Hamlin Air Conditioning Co., and (for mail), 1014 Delaware Ave., Buffalo, N. Y.
HAMLIN, **Harry A.** (*A* 1916), Br. Mgr. (for mail), Johnson Service Co., 427 Brainard St., Detroit, and 120 Winona, Highland Park, Mich.
HANLEY, **Edward V.** (*A* 1933), Pres. (for mail), S. V. Hanley Co., 1653 N. Farwell Ave., Milwaukee, and 844 E. Birch Ave., Whitefish Bay, Wis.
HANLEY, **Thomas F., Jr.** (*M* 1933), Pres. (for mail), Hanley & Co., 1503 S. Michigan Ave., and 4940 East End Ave., Chicago, Ill.
HANSEN, **Carl J.** (*J* 1935; *S* 1933), 273 Sheffield Rd., Lansdowne, Pa.
HANSEN, **Charles C.** (*M* 1928), Engr., 428 Prospect Sta., South Orange, N. J.
HANSON, **Leslie P.** (*J* 1935; *S* 1933), Engr., U. S. Air Cond. Corp., and (for mail), 4336-46th Ave. S., Minneapolis, Minn.
HARDING, **Louis A.*** (*M* 1911), (*Presidential Member*), (Pres., 1930; 1st Vice-Pres., 1929; 2nd Vice-Pres., 1928; Council, 1922-1931), Pres. (for mail), L. A. Harding Construction Corp., Prudential Bldg., and 85 Cleveland Ave., Buffalo, N. Y.
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HARMS, **William T.*** (*M* 1917), 1015 Vinewood Ave., Detroit, Mich.
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HARRINGTON, **Charles** (*M* 1923), 43 Indian Grove, Toronto, Ont., Canada.
HARRINGTON, **Elliott D.*** (*M* 1932; *A* 1930), Engr. (for mail), Air Cond. Dept., Commercial Engrg. Div., General Electric Co., 1 River Rd., and 1580 Wendell Ave., Schenectady, N. Y.
HARRIS, **Jesse B.** (*M* 1918), Pres. (for mail), Rose & Harris Eng., Inc., 416 Essex Bldg., and 3620 Colfax Ave. S., Minneapolis, Minn.
HART-BAKER, **Henry W.** (*M* 1918), Director (for mail), Merritt, Ltd., 8 French Bund, and 37 Rte Rene Delastre, Shanghai, China.
HART, **Harry M.*** (*M* 1912), (*Presidential Member*), (Pres., 1916; 1st Vice-Pres., 1915; Council, 1914-1917), Pres. (for mail), L. H. Prentice Co., 1048 Van Buren St., and 5409 Winthrop Ave., Chicago, Ill.
HARTMAN, **Fred Stewart** (*A* 1933), Dist. Mgr., Industrial Dept. (for mail), General Electric Co., 570 Lexington Ave., New York, N. Y., and 168 Montclair Ave., Montclair, N. J.

HARTMAN, John M. (*M* 1927), Engr. (for mail), Kewanee Boiler Corp., and 719 Henry St., Kewanee, Ill.

HARTWEIN, Charles E. (*M* 1933), Supervisor, House Htg. Dept., St. Louis County Gas Co., 231 W. Lockwood, Webster Groves, and (for mail), 6271 Magnolia Ave., St. Louis, Mo.

HARTWELL, Joseph C. (*M* 1922), (for mail), Hartwell Co., Inc., 87 Weybosset St., and 16 Freeman Pkwy., Providence, R. I.

HARVEY, Alexander D. (*A* 1928; *J* 1925), Nash Engrg. Co., South Norwalk, Conn.

HARVEY, Lyle C. (*M* 1928), Vice-Pres. (for mail), Bryant Heater & Mfg. Co., 17825 St. Clair Ave., and 3388 Glencarin Rd., Cleveland, Ohio.

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HASLETT, Henry M. (*S* 1935), 950 Lombard Ave., St. Paul, Minn.

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HAUAN, Merlin J. (*M* 1933), Consulting Engr., 3412–16th S., Seattle, Wash.

HAUPT, Howard F. (*A* 1929), 614 E. Beaumont Ave., Milwaukee, Wis.

HAUSS, Charles F.* (*Charter Member; Life Member*), Via Gioberti No. 2, Milano, Italy.

HAYDEN, Carl F. (*A* 1930), Br. Mgr. (for mail), Barber-Colman Co., 221 N. LaSalle St., Chicago, and 2227 Ewing Ave., Evanston, Ill.

HAYES, James J. (*M* 1920), Sales Engr. (for mail), Stannard Power Equipment Co., 53 W. Jackson Blvd., Room 925, and 7443 Jeffery Ave., Chicago, Ill.

HAYES, John J. (*A* 1933), Auburn Stoker Sales Corp., 406 N. Wells St., Chicago, and (for mail), 918 Michigan Ave., Evanston, Ill.

HAYES, Joseph G. (*M* 1908), Pres. and Engr. (for mail), Hayes Bros., Inc., 236 W. Vermont St., and 2849 N. Capitol Ave., Indianapolis, Ind.

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HAYNES, Charles V. (*M* 1917), (*Presidential Member*), (Pres., 1934; 1st Vice-Pres., 1933; 2nd Vice-Pres., 1932; Council, 1926-1929, 1932-1934), Vice-Pres., Hoffman Specialty Co., 500 Fifth Ave., Room 3324, New York, N. Y., and (for mail), 115 Llanfair Rd., Ardmore, Mont. Co., Pa.

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HEARD, John A. E. (*J* 1930), Carrier Engrg. Co., Ltd., Sardar Sujan Singh Block, Connaught Pl., New Delhi, India, and (for mail), "Lynton" 28 Leighcliff Rd., Leighon-Sea, Essex, England.

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HEATH, William R. (*M* 1931), Asst. Chief Engr., Buffalo Forge Co., 490 Broadway, and (for mail), 119 Wingate Ave., Buffalo, N. Y.

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HESTER, Thomas J. (*M* 1919), Vice-Pres-Treas. (for mail), Hester-Bradley Co., 2835 Washington Blvd., and 67 Aberdeen Pl., St. Louis, Mo.

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HILL, Fred M. (*M* 1930), 225 East Ave. 39, Los Angeles, Calif.
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PLASS, Charles Webster (*M* 1928), 826 E. Haines St., Philadelphia, Pa.

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PLUNKETT, John H. (*M* 1925), 81 Woodrow Ave., Boston, Mass.

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PRESDEE, Cliff W. (*A* 1926), Adv. Mgr., Heating & Ventilating, 148 Lafayette St., New York, N. Y.
PRICE, Charles E. (*A* 1933), Treas. (for mail), Keeney Publishing Co., 6 N. Michigan Ave., Chicago, and 1151 Chatfield Rd., Winnetka, Ill.
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ROTH, Charles F. (*A* 1930), Mgr., International Htg. and Vtg. Exposition, Grand Central Palace, New York, and (for mail), 141 East 36th St., New York (November 1 to April 30), and Dreamthorp, Bedford Village, N. Y. (May 1 to October 31).

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RUSSELL, William B. (*M* 1928), Colorado Ave., R. F. D. No. 1., Joliet, Ill.

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SCOTT, William P., Jr. (*J* 1933), 9 Scenic Way, San Francisco, Calif.

SCRIBNER, Eugene D. (*A* 1933; *J* 1929), Engr. (for mail), Carrier Engrg. Corp., Chrysler Bldg., R. 408 New York, N. Y., and 264 Prospect St., Westfield, N. J.

SEEBER, Rex R.* (*M* 1934), Head, Mech. Engrg. Dept., Michigan College of Mining and Technology, Houghton, Mich.

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SEELEY, Lauren E.* (*M* 1930), Asst. Prof. (for mail), Yale Engrg. School, Yale University, Mason Lab., and 130 Everit St., New Haven, Conn.

SEELIG, Alfred E. (*M* 1926), Pres. and Gen. Mgr., L. J. Wing Mfg. Co., 154 West 14th St., and (for mail), 310 Convent Ave., New York, N. Y.

SEELIG, Lester (*M* 1925), Mech. Engr., Museum of Science and Industry, Jackson Park, and (for mail), 725 Irving Park Blvd., Chicago, Ill.

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SEKIDO, Kunisuke (*M* 1903), Consulting Engr., Marunouchi Bldg., No. 855, and (for mail), 19 Momozono Nakano, Tokyo, Japan.

SELLMAN, Nils T. (*M* 1922), Director of Sales and Utilization, and Asst. Secy. (for mail), Consolidated Gas Co. of New York, 4 Irving Pl., and 56 Walworth Ave., Scarsdale, N. Y.

SENIOR, Richard L. (*M* 1925), (for mail), R. L. Senior, Inc., 103 Park Ave., New York, and 10 Cherry Ave., New Rochelle, N. Y.

SENNET, Lowell E. (*S* 1934), 1719 East 115th St., Cleveland, Ohio.

SEVERNS, William H.* (*M* 1933), Prof. of Mech. Engr. (for mail), Dept. of Mech. Engrg., University of Illinois, and 609 Indiana Ave., Urbana, Ill.

SEWARD, Percival H.* (*Charter Member; Life Member*), Research, 369 Washington Ave., Brooklyn, N. Y.

SHAER, I. Ernest (*A* 1934), Sales Engr., B. F. Sturtevant Co., 80 Broad St., Boston, and (for mail), 35 Fessenden St., Dorchester, Mass.

SHANKLIN, Arthur P. (*M* 1920), Sales Engr. (for mail), Carrier Engrg. Corp., 12 South 12th St., Philadelphia, and 40 Amherst Ave., Swarthmore, Pa.

SHANKLIN, John A. (*M* 1928), Secy-Treas. (for mail), West Virginia Htg. & Plbg. Co., 233 Hale St., and 1507 Quarrier St., Charleston, W. Va.

SHARP, Floyd H. (*M* 1929), 117 E. Third St., Jamestown, N. Y.

SHARP, Henry C. (*M* 1935), Mgr., Oil Heat Div. (for mail), Smith Oil & Refining Co., 1104 Kilburn Ave., and 1928 Rockton Ave., Rockford, Ill.

SHAVER, Herbert H. (*A* 1929), Asst. Gen. Sales Agent (for mail), Hudson Coal Co., 424 Wyoming Ave., and 1507 Wyoming Ave., Scranton, Pa.

SHAW, Burton E. (*J* 1934), Research Chief, Gilbert & Barker Mfg. Co., Springfield, and (for mail), Granby Rd., Southwick, Mass.

SHAW, Edgar (*M* 1923), Pres. (for mail), Lynch & Woodward, Inc., 320 Dover St., Boston, and 51 Royal St., Wollaston, Mass.

SHAW, Harold Wilson (*S* 1935), 815 Fairmount Ave., St. Paul, Minn.

SHAW, Norman J. H. (*M* 1927; *J* 1925), 37 Benjamin Rd., Arlington, Mass.

SHAWLIN, Walter C. (*A* 1931), 696 S. Oak Park Court, Milwaukee, Wis.

SHEA, Michael B. (*M* 1921), Sales Dept. (for mail), American Radiator Co., 1344 Broadway, Detroit, and 114 Massachusetts Ave., Highland Park, Mich.

SHEARS, Matthew W. (*M* 1922), 39 Sylvan Ave., Toronto, Ont., Canada.

SHEFFLER, Morris (*M* 1921), Pres. (for mail), Sheffler-Gross Co., 203 Drexel Bldg., and 5451 Lebanon Ave., Philadelphia, Pa.

SHELDON, Nelson E. (*M* 1927), Dist. Sales Mgr. (for mail), Carrier Engrg. Corp., 916 Temple Bldg., and 41 Lanark Crescent, Rochester, N. Y.

SHELDON, William D., Jr. (*J* 1934), Chief Engr., Sheldons, Ltd., and (for mail), Cedar St., Galt, Ont., Canada.

SHELNEY, Thomas (*M* 1931), Pres. (for mail), Pierce Blower Corp., 27 Carolina St., and Hotel Fillmore, Buffalo, N. Y.

SHENK, Donald Hugh (*M* 1934), 106 Forest Lane, and (for mail), Riggs Hall, Clemson College, S. C.

SHEPARD, Edward C. (*M* 1932), Owner (for mail), Shepard Engrg. Co., 370 Lexington Ave., and 978 Grant Ave., New York, N. Y.

SHEPARD, John deB. (*J* 1929), Consolidated Gas, Electric Light & Power Co., Room 406 Lexington Bldg., Baltimore, Md.

SHEPPARD, Frank A. (*M* 1918), Salesman (for mail), Johnson Service Co., 411 East 10th St., and 27 East 70th St., Kansas City, Mo.

SHEPPARD, William G. F. (*M* 1922), Partner (for mail), Sheppard & Abbott, 119 Harbord St., and 1 Clarendon Ave., Toronto, Ont., Canada.

SHERET, Andrew (*M* 1929; *A* 1925), Pres. (for mail), Andrew Sheret, Ltd., 1114 Blanshard St., and 1030 St. Charles St., Victoria, B. C., Canada.

SHERMAN, Ralph A. (*M* 1933), Fuel Engr. (for mail), Battelle Memorial Institute, 505 King Ave., and 1893 Coventry Rd., Columbus, Ohio.

SHIVERS, Paul F. (*M* 1930), Chief Engr., Minneapolis-Honeywell Regulator Co., Wabash, Ind.

SHODRON, John G. (*M* 1921), Consulting Engr. and Research, 419 E. Milwaukee Ave., Ft. Atkinson, Wis.

SHORB, Will A. (*M* 1909), Treas., The Field & Shorb Co., 705 N. Pine St., and (for mail), 3 Lincoln Pl., Decatur, Ill.

SHROCK, John H. (*M* 1924), Mgr. (for mail), New York, Blower Co., Factory St., and 1524 Michigan Ave., La Porte, Ind.

SHULTZ, Earle (*A* 1919), Vice-Pres. (for mail), Illinois Maintenance Co., 1136-72 W. Adams St., and Edgewater Beach Apts., Chicago, Ill.

SIEBS, Claude T. (*A* 1927), Service Systems Engr. (for mail), Western Electric Co., Inc., 195 Broadway, New York, N. Y., and Russell Rd., Fanwood, N. J.

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SIGMUND, Ralph W. (*M* 1932), Dist. Mgr. (for mail), B. F. Sturtevant Co., 913 Provident Bank Bldg., and 304 Oak St., Cincinnati, Ohio.

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SIMONDS, Abe H. (*A* 1935; *J* 1929), Sales Engr. (for mail), Carrier Engrg. Corp. of California, 748 E. Washington Blvd., and 446 Westminster Ave., Los Angeles, Calif.

SIMPSON, Donald C. (*M* 1932), Supt. of Research, Industrial Mat. Div. (for mail), Owens, Illinois Glass Co., Newark, and 878 Kelton Ave., Columbus, Ohio.

SIMPSON, William K. (*M* 1919), Vice-Pres. (for mail), Hoffman Specialty Co., and 9 Sands St., Waterbury, Conn.

SKIDMORE, John G. (*J* 1930), Air Cond. Engr., Carrier Engrg. Corp., 408 Chrysler Bldg., New York, and (for mail), 5101-39th Ave., Long Island City, N. Y.

SKINNER, Henry W. (*M* 1920), Consulting Engr. (for mail), Box 1334, and 4816 Dexter, Ft. Worth, Texas.
SKLENÁRIK, Louis (*J* 1928), 305 East 72nd St., New York, N. Y.
SLAYTER, Games (*M* 1931), 68 Walhalla Rd., and (for mail), 711 Southwood Ave., Columbus, Ohio.
SLIGHT, Irvin (*A* 1925), Slight Bros., 741 Yorkway Pl., Jenkintown, Pa.
SMAK, Julius R. (*A* 1934), Supt. of Service Depts., Crane Co., South Ave., and (for mail), 3135 Park Ave., Bridgeport, Conn.
SMALL, Bartlett R. (*J* 1932), Sales Engr. (for mail), T. C. Heyward, 1408 Independence Bldg., and 326 West 10th St., Charlotte, N. C.
SMALL, John D.* (*M* 1910), Consulting Engr. (for mail), 127 N. Dearborn St., Chicago, and 411 Maple Ave., Wilmette, Ill.
SMALLMAN, Edwin W. (*M* 1920), Navy Dept., and (for mail), 831 Allison St. N.W., Washington, D. C.
SMITH, Elmer G.* (*M* 1929), Asst. Prof. of Physics, Agricultural and Mechanical College of Texas, College Station, Texas.
SMITH, Gard W. (*M* 1927), Salesman, Premier Warm Air Heater Co., Dowagiac, Mich., and (for mail), 248 Roche St., Huntington, Ind.
SMITH, Jared A. (*A* 1933), Br. Mgr. (for mail), The Bryant Heater & Mfg. Co., 626 Broadway, and 3817 Indian View Ave., Mariemont, Cincinnati, Ohio.
SMITH, J. Darrell (*M* 1933), Mech. Engrg. Dept., Philadelphia & Reading Coal & Iron Co., and (for mail), 317 North 19th St., Pottsville, Pa.
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SMITH, Robert Hugh (*J* 1934; *S* 1933), Sales Engr., Sears Roebuck & Co., Dept. of Plbg., Htg. and Vtg., 135 S. Fifth St., and (for mail), 214 N. Fourth St., Room 410, Steubenville, Ohio.
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SMOOT, Theo Halley (*M* 1935), Chief Engr. (for mail), Fluid Heat Div., Anchor Post Fence Co., Eastern Ave. and Kane St., and 2512 Talbot Rd., Baltimore, Md.
SMYERS, Edward C. (*A* 1933), Sales Engr., Minneapolis-Honeywell Regulator Co., 1013 Penn Ave., Wilkinsburg, and (for mail), 148 Jamaica Ave., West View, Pittsburgh, Pa.
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SNELL, Ernest (*M* 1920), 3914 LeMay Ave., Detroit, Mich.
SNIDER, Lewis A. (*M* 1927), Pres. (for mail), L. A. Snider Engrg. Service, Inc., 605 N. Michigan Ave., and 649 Buena Ave., Chicago, Ill.
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SNYDER, Jay W. (*M* 1917), McColl-Snyder-McLean, 2304 Penobscot Bldg., Detroit, Mich.
SNYDER, Joseph S. (*A* 1925), Sales Repr., Detroit Lubricator Co., 374 Delaware Ave., and (for mail), 9 Knowlton Ave., Buffalo, N. Y.
SODEMANN, Paul W. (*M* 1926; *J* 1920), Sales Engr., 2306 Delmar Blvd., and (for mail), 4136 Farlin Ave., St. Louis, Mo.
SODEMANN, William C. B. (*M* 1919), Pres. (for mail), Sodemann Heat & Power Co., 2306 Delmar Blvd., St. Louis, Mo.
SONNEBORN, Charles (*M* 1930), Vice-Pres. in charge of Production, Shaw, Perkins Mfg. Co., West Pittsburgh, and (for mail), R. D. No. 3, New Castle, Pa.
SONNEY, Kermit J. (*S* 1934), (for mail), 316 W. Symmes St., Norman, Okla., and L. B. 136, Wilcox, Pa.
SOPER, H. A. (*M* 1916), Vice-Pres., American Foundry & Furnace Co., Bloomington, Ill.
SOULE, Lawrence C.* (*M* 1908), Secy. and Chief Engr. (for mail), Aerofin Corp., 850 Frelinghuysen Ave., Newark, and Essex Fells, N. J.
SPAFFORD, Allen (*A* 1927), Wood Conversion Co., Cloquet, Minn.
SPECKMAN, Charles H. (*M* 1918), Prof. Engr., 375 Bourse Bldg., Philadelphia, Pa.
SPELLER, Frank N.* (*M* 1908), Director, Dept. of Metallurgy and Research (for mail), National Tube Co., 1922 Frick Bldg., and 6411 Darlington Rd., Pittsburgh, Pa.
SPENCE, Morton R. (*J* 1934), Asst. Purchasing Agent, Rundle & Spence Mfg. Co., 445 N. Fourth St., and (for mail), 709 E. Lexington Blvd., Milwaukee, Wis.
SPENCER, Roland M. (*J* 1934), Sales Engr. (for mail), Powers Regulator Co., 754 Hippodrome Annex, Cleveland, and 1269 Bonnie View Ave., Lakewood, Ohio.
SPIELMAN, Gordon P. (*A* 1931; *J* 1923), Harrison, Spielman Co., 480 Milwaukee Ave., Chicago, Ill.
SPIELMANN, Harold J. (*M* 1933), Air Cond. Engr., The Vilter Mfg. Co., and (for mail), 2549 N. Lake Dr., Milwaukee, Wis.
SPITZLEY, Ray L. (*M* 1920), 1200 W. Fort St., Detroit, Mich.
SPOFFORTH, Walter (*M* 1930), Chief of Mech. Services, U. S. Penitentiary, McNeil Island, and (for mail), 1850 W. Blvd., Day Island, Tacoma, Wash.
SPROULL, Howard E. (*M* 1920), Div. Sales Mgr. (for mail), American Blower Corp., 1005-6 American Bldg., and 3588 Raymar Dr., Cincinnati, Ohio.
SPURGEON, Joseph H. (*M* 1924), Salesman (for mail), Spurgeon Co., 5-203 General Motors Bldg., and 17215 Pennington Dr., Detroit, Mich.
STACEY, Alfred E., Jr.* (*M* 1914), Wooton Rd., Essex Fells, N. J.
STACK, Frank Charles (*J* 1935; *S* 1933), 140-23 Cherry Ave., Flushing, N. Y.
STACY, Stanley C. (*M* 1931), Mech. Engr. (for mail), Board of Education, 13 S. Fitzhugh St., and 91 Cobbs Hill Dr., Rochester, N. Y.
STALB, Joseph G. (*A* 1934), New York Mgr., Parco Furnace Div., Reading Iron Co., 143 Liberty St., and (for mail), 22 Partridge Ave., Ridley Park, Pa.
STAMMER, Edward L. (*M* 1919), Supt., Htg. and Vtg., Board of Education Bldg., and (for mail), 4430 Tennesee Ave., St. Louis, Mo.
STANGER, Ralph B. (*M* 1920), Owner (for mail), Robinson & Stanger, Empire Bldg., Pittsburgh, and Deer Creek, Church Rd., Glenshaw, Pa.
STANGLAND, B. F. (*Charter Member*), (2nd Vice-Pres., 1908; Board of Governors, 1905, 1906, 1909; Board of Mgrs., 1895-1899; Council, 1896-1897), Kendall, N. Y.
STANNARD, James M.* (*Life Member; M* 1906), Pres-Treas. (for mail), Stannard Power Equipment Co., 53 W. Jackson Blvd., Chicago, and 1402 Elinor Pl., Evanston, Ill.
STAPLES, William H. (*A* 1924), Htg. and Vtg. (for mail), Maguire Staples & Mason, 24 West 20th St., and 548 West 164th St., New York, N. Y.
STARK, W. Elliott* (*M* 1926), (Council, 1932-1934), Research Engr., Bryant Heater & Mfg. Co., 17825 St. Clair Ave., Cleveland, and (for mail), 1875 Rosemont Rd., East Cleveland, Ohio.
STEELE, John B. (*M* 1932), Chief Engr. (for mail), Engrg. Dept., Winnipeg School Board, Ellen and William Ave., and 184 Waterloo St., Winnipeg, Man., Canada.
STEELE, Maurice G. (*M* 1929), Product Engr. (for mail), Revere Copper & Brass, Inc., Research Dept., and 906 N. Madison St., Rome, N. Y.
STEEN, Joseph M. (*M* 1929), Iron City Htg. Co., 843 Jacksonia St., Pittsburgh, Pa.
STEFFNER, Edward F. (*J* 1934), 10517 Fortune Ave., Cleveland, Ohio.

STEGGALL, Howard B. (*A* 1934), Br. Mgr. (for mail), U. S. Radiator Corp., 941 Behan St., and 1166 Murray Hill Ave., Pittsburgh, Pa.

STEINHORST, Theodore F. (*M* 1919), Treas. and Gen. Mgr., Emil Steinhorst & Sons, Inc., 612 South St., and (for mail), 1664 Brinckerhoff Ave., Utica, N. Y.

STEINKELLNER, Edward J. (*S* 1935), 315–19th Ave. S.E., Minneapolis, Minn.

STEINMETZ, C. W. Arthur (*M* 1934), Mgr. (for mail), American Blower Corp., 402 Broad St., Newark, and 50 Oakwood Ave., Bogota, N. J.

STEPHENSON, L. A. (*M* 1917), Mgr. (for mail), Powers Regulator Co., 409 East 13th St., and 801 West 57th Terrace, Kansas City, Mo.

STERNBERG, Edwin (*A* 1932; *J* 1931), Air Cond. Engr., Arctic Engrg. Co., 123 White St., and (for mail), 58 East 92nd St., New York, N. Y.

STERNE, Cecil M. (*A* 1934), Chief Engr. (for mail), Metropolitan Refining Co., Inc., 23-28 50th Ave., Long Island City, and 115 Harold Rd., Woodmere, L. I., N. Y.

STETSON, Lawrence R. (*M* 1913), 303 Congress St., Boston, Mass.

STEVENS, Harry L. (*M* 1934; *A* 1927; *J* 1924), Secy-Treas. (for mail), M. M. Stevens Co., 108 W. Sherman, and 7 West 22nd St., Hutchinson, Kans.

STEVENS, John M. (*A* 1933), 4643 Morris St., Philadelphia, Pa.

STEVENS, William R. (*A* 1934), Partner, L. E. Stevens Co., 442 E. Front St., Cincinnati, Ohio, and (for mail), 159 Tremont Ave., Ft. Thomas, Ky.

STEVENSON, Wilbur W. (*M* 1928), Steam Htg. Engr. (for mail), Allegheny County Steam Htg. Co., 435 Sixth Ave., and 1125 Lancaster Ave., Pittsburgh, Pa.

STEWART, Charles W. (*M* 1919; *A* 1918), Asst. Secy. (for mail), Hoffman Specialty Co., and 21 Yates Ave., Waterbury, Conn.

STEWART, Clement W. (*M* 1934), Sales Engr. (for mail), Ilg Electric Vtg. Co., 15 Park Row., and 3985 Saxon Ave., New York, N. Y.

STEWART, Duncan J. (*A* 1930), Mgr., Electric Apparatus Div. (for mail), Barber-Colman Co., and 214 Franklin Pl., Rockford, Ill.

STEWART, John C. (*A* 1934), Owner (for mail), 1844 Smith St., and 2807 Victoria Ave., Regina Sask, Canada.

STILL, Fred R.* (*M* 1904), (*Presidential Member*), (Pres., 1918; 2nd Vice-Pres., 1917; Council, 1916-1919), Vice-Pres. (for mail), American Blower Corp., 401 Broadway, and 1 East End Ave., New York, N. Y.

STILLER, Frederick Wilbur (*J* 1933), Estimator (for mail), F. C. Stiller & Co., 129 S. Tenth St., and 138 West 49th St., Minneapolis, Minn.

STINARD, Rutherford L. (*J* 1934), Engr., American Radiator Co., 40 West 40th St., New York, N. Y., and (for mail), 1377 Boulevard E, West New York, N. J.

STITT, Arthur B. (*J* 1935; *S* 1933), Plbg. and Htg. Engr., Sears Roebuck & Co., 184 Atlantic St., Stamford, Conn., and (for mail), 260 Valentine Lane, Yonkers, N. Y.

STITT, Eugene W. (*M* 1917), Sales Repr., Cast Products Div. (for mail), 45 Second Ave., Johnstown, Pa.

STOCKWELL, William R. (*M* 1903; *J* 1901), Weil-McLain Co., Michigan City, Ind.

STONE, Eugene R. (*M* 1913), 78 Woodbine St., Quincy, Mass.

STONE, George F. (*Life Member*; *M* 1918), Estimator, 16 Elmwood Rd., Verona, N. J.

STRAUCH, Paul C. (*A* 1934), Sales Engr., The Henry Furnace & Foundry Co., 18th and Merriman Sts., Pittsburgh, and (for mail), 101 Washington Ave., Edgewood, Pittsburgh, Pa.

STREVELL, Roger P. (*M* 1934), Co-Partner (for mail), Wm. R. Hogg Co., 900 Fourth Ave., Asbury Park, and corner State Highway and Victor Pl., Neptune, N. J.

STRICKLAND, Albert W. (*A* 1929), Htg. and Vtg. Engr., Big Timber, Mont.

STROCK, Clifford (*A* 1929), Associate Editor (for mail), Heating and Ventilating, 148 Lafayette St., and 150 East 182nd St., New York, N. Y.

STROUSE, Sherman W. (*A* 1934), Sales Engr., Cooney Refrigeration Co., Inc., and (for mail), 315 Capen Blvd., Buffalo, N. Y.

STROUSE, Sidney B. (*M* 1921), Engr. (for mail), 500-529 Guarantee Trust Bldg., and 22 S. Illinois Ave., Atlantic City, N. J.

STRUNIN, Jay (*J* 1933), Engr. and Contractor (for mail), Strunin Plbg. & Htg. Co., 408 Second Ave., and 54 West 89th St., New York, N. Y.

STUBBS, W. C. (*M* 1934), Design Draftsman, Heat and Vtg. (for mail), Norfolk Navy Yard, and 36 Channing Ave., Portsmouth, Va.

SUMMERS, Ernest T. (*A* 1930), Pres. (for mail), Summers, Darling & Co., 121 Smith St., and Ste. 22 Newcastle Apts., Winnipeg, Man., Canada.

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SUTCLIFFE, Arthur G. (*M* 1922; *A* 1918), Chief Engr., Ilg Electric Vtg. Co., 2850 N. Crawford Ave., and (for mail), 4146 N. St. Louis Ave., Chicago, Ill.

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SWANEY, Carroll R. (*M* 1929; *J* 1921), Gilbert Howe Gleason, 25 Huntington Ave., Boston, Mass.

SWANSON, Harry (*M* 1933), Engr. (for mail), The Fels Co., 42 Union St., Portland, and Box 135, Cape Cottage, Maine.

SWANSON, Rolf G. (*S* 1935), 324 Walnut St. S.E., Minneapolis, Minn.

SWANSTROM, Alfred E. (*J* 1935; *S* 1932), Construction Foreman, U. S. Dept. of Interior, and (for mail), 1444 Van Buren St., St. Paul, Minn.

SWEATT, Charles H. (*S* 1935), 4259 Unity Ave., Robbinsdale, Minn.

SWEIVEN, C. E. (*S* 1935), 406 Walnut St. S.E., Minneapolis, Minn.

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SWISHER, Stephen G., Jr. (*A* 1934), Sales Engr. (for mail), The Trane Co., 125 E. Wells St., and 4238 N. Woodburn Ave., Milwaukee, Wis.

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SZEKELY, Ernest (*M* 1920), Vice-Pres. and Gen. Mgr. (for mail), Bayley Blower Co., 1817 South 66th St., and 3104 W. Kilbourn Ave., Milwaukee, Wis.

SZOMBATHY, Louis R. (*A* 1930), Ferguson Sheet Metal Works, Inc., 34 N. Florissant Blvd., Ferguson, Mo.

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TABOR, Charles B. (*S* 1935), 1815 University Ave. S.E., Minneapolis, Minn.

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TALLMADGE, Webster (*M* 1924), Pres. (for mail), Webster Tallmadge & Co., Inc., 255 North 18th St., East Orange, and 7 Claremont Pl., Montclair, N. J.

TARR, Harold M. (*M* 1931), Htg. Engr., 21 Montague St., Arlington Heights, Mass.
TAUSON, Peter O. (*S* 1934), 408 Southwest 23rd St., Oklahoma City, Okla.
TAVANLAR, Eligio J.* (*J* 1931), (for mail), Carrier Research Corp., 850 Frelinghuysen Ave., Newark, N. J., and Binalonan Pangasinan, Philippine Islands.
TAVERNA, Frederick F. (*M* 1928; *A* 1927; *J* 1924), Engr., Raisler Htg. Co., 129 Amsterdam Ave., New York, N. Y., and (for mail), 406-12th St., Union City, N. J.
TAYLOR, Edward M. (*A* 1934), Draughtsman and Engrg. Asst., City Engineer's Dept., and (for mail), 102 Innes Rd., Christchurch, New Zealand.
TAYLOR, William E. (*A* 1934), Factory Sales Engr., Air Cond. Div. of (for mail), Gar Wood Industries, Inc., 7924 Riopelle St., and 2008 W. Grand Blvd., Detroit, Mich.
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TEELING, George A. (*M* 1930), Consulting Engr., 11 N. Pearl St., Albany, N. Y.
TEMPLE, Walter J. (*M* 1931), Engr., J. A. Temple & Co., 919 E. Michigan Ave., and (for mail), 1215 Reed St., Kalamazoo, Mich.
TEMPLIN, Charles L. (*M* 1921), Sales Engr. (for mail), Carrier Engrg. Corp., Bona Allen Bldg., and 781 Sherwood Rd. N.E., Atlanta, Ga.
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TENNANT, Raymond J. J. (*A* 1929), Supervisor of Sales (for mail), Duquesne Light Co., 435 Sixth Ave., and 529 Navato Pl., Pittsburgh, Pa.
TENNEY, Dwight (*M* 1932), Pres. and Chief Engr. (for mail), Tenney Engrg., Inc., Bloomfield, Ave., at Grove St., Bloomfield, and 33 Summit Rd., Verona, N. J.
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THOMAS, L. G. Lee (*M* 1934), Vice-Pres. (for mail), Economy Pumping Machinery Co., 3431 West 48th Pl., Chicago, and 426 Forest Ave., Oak Park, Ill.
THOMAS, Melvern F. (*M* 1909), Consulting Engr. (for mail), Thomas & Wardell, 229 College St., and 24 Rivercrest Rd., Toronto, Ont., Canada.
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THOMPSON, Nelson S.* (*M* 1917; *J* 1897), 1615 Hobart St. N.W., Washington, D. C.
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THORNTON, Roger T. (*M* 1919), Buffalo Forge Co., 490 Broadway, Buffalo, N. Y.
THORNTON, William B.* (*M* 1931), Sales Engr. (for mail), Carrier Engrg. Corp., 404 Bona Allen Bldg., Atlanta, and 155 Coventry Rd., Decatur, Ga.
THRUSH, Homer A. (*M* 1918), H. A. Thrush & Co., 21-23 E. Riverside Dr., Peru, Ind.
TIBBETS, John C. (*M* 1920), Engrg. Dept., B. & O. R. R. Co., and (for mail), P.O. Box 106, Ellicott City, Md.
TILLER, Louin (*J* 1935; *S* 1933), 1724 Northwest 20th St., Oklahoma City, Okla.
TILTZ, Bernard E. (*M* 1930), Pres. (for mail), Tiltz Air Cond. Corp., 285 Madison Ave., New York, and 24 Barnum Rd., Larchmont, N. Y.
TIMMIS, Pierce (*M* 1920), Service Equip. Dept. (for mail), United Engineers & Constructors, Inc., 1401 Arch St., Philadelphia, and 202 Midland Ave., Wayne, Pa.
TIMMIS, W. Walter (*M* 1933; *A* 1925), Engr. (for mail), American Radiator Co., 40 West 40th St., New York, and 32 Oak Lane, Glen Cove., N. Y.
TISNOWER, William (*M* 1923), 131 Livingston St., Brooklyn, N. Y.
TITUS, Marvin S. (*M* 1928), 414 Fayette St., Charleston, W. Va.
TJERSLAND, Alf (*M* 1916; *J* 1906), E. Sunde & Co., Ltd., Oslo, Norway.
TOBIN, George J. (*M* 1905), Owner, Sanitary, Htg. & Vtg. Engr., 187 North Ave., Plainfield, N. J.
TOBIN, John F. (*A* 1934), Salesman, American Blower Corp., Rm. 1404, 228 N. LaSalle St., Chicago, Ill.
TOONDER, Clarence L. (*M* 1933), Air Cond. Engr., Sales Engrg. Dept., Kelvinator Corp., Plymouth Rd., and (for mail), 12761 Stratmoor Ave., Detroit, Mich.
TORNQUIST, Earl L. (*A* 1934), Supervisor, Distribution Operation (for mail), Public Service Co. of North Illinois, 72 W. Adams St., Chicago, and 465 Parkside Ave., Elmhurst, Ill.
TORR, Thomas W. (*M* 1933), Chief Engr., The Rudy Furnace Co., P. O. Box 73, Dowagiac, Mich.
TORRANCE, Henry (*M* 1933), Pres., 175 Christopher St., and (for mail), 112 East 17th St., New York, N. Y.
TOUTON, R. D. (*M* 1933), Tech. Director (for mail), Bayuk Cigars, Inc., Ninth and Columbia Ave., Philadelphia, and 19 Lodges Lane, Cynwyd, Pa.
TOWER, Elwood S. (*M* 1930), Engr., 1114 Koppers Bldg., and (for mail), 1411 Wightman St., Pittsburgh, Pa.
TRANE, Reuben N.* (*M* 1915), Pres. (for mail), The Trane Co., and 126 South 15th St., LaCrosse, Wis.
TRAUGOTT, Mortimer (*A* 1930), East Sales Mgr. (for mail), Bryant Heater & Mfg. Co., 152 North 15th St., Philadelphia, and 721 Meeting House Rd., Elkins Park, Pa.
TREADWAY, Quentin (*J* 1932), Sales Engr. (for mail), Clarage Fan Co., 707 Security Bank Bldg., and 2618 Collingwood, Toledo, Ohio.
TRIMMER, Charles M. (*J* 1935; *S* 1933), Inspection and Testing, Rockland Light & Power Co., 105 Pike St., and (for mail), 28 Prospect St., Port Jervis, N. Y.
TROSKE, Joseph J. (*A* 1931), Vice-Pres. and Gen. Mgr. (for mail), Vander-Troske Co., 236 Winter Ave. N.W., and 233 Brown St. S.E., Grand Rapids, Mich.
TRUITT, Joseph E. (*M* 1920; *A* 1911), Pres., Autovent Fan & Blower Co., 1805 N. Kostner Ave., Chicago, Ill.
TRULSON, Arthur F. (*M* 1930), Mech. Engr., 1509 W. Sixth St., Ashland, Wis.
TRUMBO, Silas M. (*A* 1926), Sales (for mail), Buffalo Forge Co., 20 N. Wacker Dr., Chicago, and 921 Franklin St., Downers Grove, Ill.
TRUMP, Charles C. (*M* 1934), Pres. and M. E. (for mail), James Spear Stove & Htg. Co., 1823 Market St., Philadelphia, and 503 Baird Rd., Merion, Pa.
TUCKER, Frank N. (*M* 1926), Field Engr., Ilg Electric Vtg. Co., Room 1108, 13 Park Row, New York, and (for mail), 289 Whaley St., Freeport, L. I., N. Y.

TUCKERMAN, George E. (*M* 1932), Mgr. Philadelphia Br., Air Cond. Div. (for mail), York Ice Machinery Corp., 1238 North 44th St., and 6202 Ogontz Ave., Philadelphia, Pa.

TURLAND, Charles H. (*M* 1934; *A* 1930), Mgr., Htg. and Vtg. Dept., Kipp-Kelly, Ltd., 68 Higgins Ave., and (for mail), 325 Centennial St., Winnipeg, Man., Canada.

TURNAU, Edmund H. (*J* 1935; *S* 1933), Cadet Engr., Koppers Seaboard By-Product Coke Co., and (for mail), 23 Polifly Rd., Hackensack, N. J.

TURNER, George G. (*A* 1934), Western Repr. (for mail), Heating and Ventilating, 228 N. LaSalle St., Chicago, and 803 Elmwood Ave., Evanston, Ill.

TURNER, John (*M* 1930), Sales Engr. (for mail), Minneapolis-Honeywell Regulator Co., 285 Columbus Ave., Boston, Mass., and Contoocook, N. H.

TURNER, John W. (*M* 1928), Chief Engr. (for mail), Pacific Steel Boiler Div., Box 1488, Detroit, and 26031 Concord Rd., Royal Oak, Mich.

TURNER, Mebane E. (*M* 1934), Mech. Engr., R. J. Reynolds Tobacco Co., and (for mail), 643 Holly Ave., Winston Salem, N. C.

TURNO, Walter G. W. (*M* 1917; *A* 1912), Secy., H. W. Porter & Co., Newark, and (for mail), 71 Lafayette Ave., East Orange, N. J.

TUSCH, Walter (*M* 1917), Htg. and Vtg. Engr., Tenney & Ohmes, Inc., 101 Park Ave., New York, and (for mail), 881 Sterling Pl., Brooklyn, N. Y.

TUTTLE, George H.* (*J* 1934), Htg. Engr. (for mail), The Detroit Edison Co., 2000 Second Ave., and 9820 Belle Terre, Detroit, Mich.

TUTTLE, J. Frank (*M* 1913), Sales Agent (for mail), Warren Webster Co., Kewanee Boiler Corp., 127 Federal St., Boston, and 2 Elmwood Ave., Winchester, Mass.

TUVE, George L.* (*M* 1932), Asso. Prof. of Mech. Engrg. (for mail), Case School of Applied Science, and 1204 Cleveland Heights Blvd., Cleveland, Ohio.

TWIST, Charles F. (*M* 1921), Secy. (for mail), Ashwell-Twist Co., 967 Thomas St., and 2310 Tenth Ave. N., Seattle, Wash.

TYLER, Roy D. (*M* 1928), East Sales Mgr. (for mail), Modine Mfg. Co., 101 Park Ave., New York, and 15 Highbrook Ave., Pelham, N. Y.

TYSON, William H. (*M* 1928), Mgr. of Engrg. (for mail), Goodyear Tyre & Rubber Co., Ltd., and "Kipewa" Codsall Rd. N.R., Wolverhampton, England.

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UHLHORN, W. J. (*M* 1920), 733 S. Highland Ave., Oak Park, Ill.

ULLMAN, Herbert G.* (*A* 1928), Mgr. Mech. Product Development Lab., American Radiator Co., P. O. Box 356, Second St., Beechwood Ave., New Rochelle, and (for mail), 107 White Rd., Scarsdale, N. Y.

URDAHL, Thomas H. (*M* 1930), Consulting Engr. (for mail), 726 Jackson Pl. N.W., and 1505–44th St. N.W., Washington, D. C.

V

VALE, Henry A. L. (*M* 1929), Managing Director (for mail), Vale Co., Ltd., 141-43 Armagh St., Christchurch, and 241 Ilam Rd., Fendalton, Christchurch, New Zealand.

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VAN ALSBURG, Jerold H. (*M* 1931), Engr., Hart & Cooley Mfg. Co., and (for mail), R. No. 3, Holland, Mich.

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VANDERHOOF, Austin L. (*A* 1933), (for mail), A. L. Vanderhoof, Inc., 2341 Carnegie Ave., Cleveland, and 3120 Yorkshire Rd., Cleveland Heights, Ohio.

VAN HORN, Howard T. (*A* 1933), Dist. Mgr., Detroit Stoker Co., 1217 McKnight Bldg., and (for mail), 4537 Grand Ave., Minneapolis, Minn.

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VERNON, J. Rexford (*M* 1928; *A* 1926), (for mail), Johnson Service Co., 1355 Washington Blvd., Chicago, and 1020 Austin St., Evanston, Ill.

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VINCENT, Paul J. (*M* 1931), Paul J. Vincent Co., 2133 Maryland Ave., and (for mail), 3807 Beech Ave., Baltimore, Md.

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VOGEL, Andrew (*M* 1926), Engr. (for mail), General Electric Co., and 1821 Lenox Rd., Schenectady, N. Y.

VOGELBACH, Oscar (*M* 1923), 23 William St., North Arlington, N. J.

VOGT, John H. (*A* 1925), Mech. Engr. (for mail), New York State Dept of Labor, 80 Centre St., New York, and 87 Grant Ave., Brooklyn, N. Y.

VOGT, Joseph B. (*M* 1933; *A* 1933; *J* 1929), 1304 Grayton Rd., Grosse Point Park, Mich.

VOISINET, Walter E. (*M* 1930), Sales Repr. (for mail), Buckeye Blower Co., 250 Delaware Ave., Buffalo, and 151 Warren Ave., Kenmore, N. Y.

VOLK, Joseph H. (*M* 1923), Pres. and Treas. (for mail), Thos. E. Hoye Htg. Co., 1906 W. St. Paul Ave., and 2965 South 43rd St., Milwaukee, Wis.

VROOME, Albert E. (*M* 1932), Engr., E. I. duPont deNemours & Co., duPont Bldg., Wilmington, Del., and (for mail), 412 Morton Ave., Rutledge, Pa.

W

WACHS, Louis J. (*J* 1930), Engr., Carrier Engrg. Corp., Chrysler Bldg., New York, and (for mail), 354 East 21st St., Brooklyn, N. Y.

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WAGNER, A. M. (*A* 1921), Mgr. (for mail), American Radiator Co., 1741 W. St. Paul Ave., and 1857 N. Prospect Ave., Milwaukee, Wis.

WAGNER, Frederick H., Jr. (*M* 1934), Mgr. Air Cond. Dept., New York Office (for mail), American Blower Corp., 401 Broadway, New York, and 1126 Post Rd., Scarsdale, N. Y.

WAITE, Harry (*A* 1929), 1409 North 17th St., Superior, Wis.

WALDON, Charles D. (*A* 1932), Consulting Engr., Spencer Foundry Co., Penetang, Ont., and (for mail), 32 Ferndale Ave., Toronto, Ont., Canada.

WALKER, Alexander (*A* 1925), Br. Mgr. (for mail), C. A. Dunham Co., Ltd., 1307 Fifth St. W., and 603–13th Ave. W., Calgary, Alberta, Canada.

WALKER, Edmund R. (*M* 1934), Sales Mgr., Htg. Div. (for mail), Fedders Mfg. Co., Inc., 57 Tonawanda St., and 696 Crescent Ave., Buffalo, N. Y.

WALKER, J. Herbert* (*M* 1916), Supt. of Central Htg. (for mail), The Detroit Edison Co., 2000 Second Ave., Detroit, and 432 Arlington Rd., Birmingham, Mich.

WALKER, William Kirby (*M* 1935), Development Engr., American Radiator Co., 40 West 40th St., New York, N. Y.

WALLACE, George J. (*M* 1923), Principal 96-19-35th Ave., Corona, and (for mail), 27-36 Ericsson St., East Elmhurst, N. Y.

WALLACE, James Bee (*A* 1935), Dist. Repr., Taco Heaters, Inc., New York, N. Y., and (for mail), 16921 Sorrento Ave., Detroit, Mich.

WALLACE, Kenneth S. (*M* 1931), Htg. Engr., Peoples Gas Light & Coke Co., 1520 Milwaukee Ave., and (for mail), 5737 Kenmore Ave., Chicago, Ill.

WALLACE, William M, II (*M* 1929), Air Cond. and Htg. Contractor, 192 Lexington Ave., New York, and (for mail), 8908–196th St., Hollis, L. I., N. Y.

WALLICH, A. C. (*M* 1919), Wallich Ice Machine Co., 517 E. Larned St., and (for mail), 1667 Burlingame Ave., Detroit, Mich.

WALSH, James A. (*A* 1932; *J* 1929), Pres. and Mgr. (for mail), James A. Walsh, Inc., Architects Bldg., Philadelphia, and Gwynedd Valley, Pa.

WALSH, J. Lee (*A* 1934), Sales Mgr. and Engr., May Oil Burner Corp., Maryland and Oliver Ave., and (for mail), Temple Ct. Apts., 34th and Guilford Ave., Baltimore, Md.

WALTERS, Arthur L. (*M* 1926; *A* 1925; *J* 1924), 7284 Richmond Pl., Maplewood, Mo.

WALTERS, William T. (*M* 1917), Engr., Illinois Engrg. Co., Corner 21st St. and Racine Ave., and (for mail), 7965 Phillips Ave., Chicago, Ill.

WALTHER, Vernon H. (*M* 1928; *J* 1925), Mech. Engr., 6821 Osceola Ave., Edison Park, Chicago, Ill.

WALTERTHUM, John J. (*A* 1922), Htg. Contractor, 173 East 62nd St., New York, N. Y., and (for mail), 42-A Van Reipen Ave., Jersey City, N. J.

WALTON, Charles W., Jr. (*M* 1934), Mech. Engr. (for mail), Rockefeller Center, Inc., 30 Rockefeller Plaza, New York, N. Y., and 120 Monte Vista Ave., Ridgewood, N. J.

WANDLESS, Franklin W. (*M* 1925), Registered Engr. (for mail), 1518 Fairmount Ave., Philadelphia, and Berwyn, Pa.

WARD, Frank James (*M* 1935), The Frank J. Ward Co., Cold Spring, Ky.

WARD, Oscar G. (*M* 1919), Dist. Repr. (for mail), Johnson Service Co., 1230 California St., and 1607 Jasmine St., Denver, Colo.

WARING, J. M. S. (*M* 1932), Consulting Engr. (for mail), Chase & Waring, 17 East 42nd St., and 277 Park Ave., New York, N. Y.

WARREN, Clarence N. (*M* 1919), Vice-Pres., Hayes Bros., Inc., 236 W. Vermont St., and (for mail), 419 East 48th St., Indianapolis, Ind.

WARREN, Francis C. (*M* 1934), Salesman (for mail), American Blower Corp., 228 N. LaSalle St., Chicago, and 127 East Ave., Park Ridge, Ill.

WARREN, Harry L. (*M* 1930), 1303 Huntington Dr., South Pasadena, Calif.

WASHBURN, Marcus J. (*A* 1934), Insulation Engr. (for mail), Eagle-Picher Lead Co., Temple Bar Bldg., and 2211 Park Ave., Cincinnati, Ohio.

WASHINGTON, George (*M* 1934), Sales Engr. (for mail), Hoffman Specialty Co., 130 N. Wells St., Chicago, and 4327 Johnson Ave., Western Springs, Ill.

WASHINGTON, Laurence W. (*M* 1929), 2301 Knox Ave., Chicago, Ill.

WATERMAN, John H. (*M* 1931), Engr. (for mail), Chas. T. Main, Inc., 201 Devonshire St., Boston, and 7 Centre St., Cambridge, Mass.

WATERS, George G. (*M* 1931; *A* 1926), Dist. Mgr. (for mail), American Blower Corp., 1433 Oliver Bldg., and 52 Vernon Dr., Pittsburgh (16), Pa.

WATSON, M. Barry (*M* 1928), Consulting Engr., 121 Welland Ave., Toronto 5, Canada.

WAUNG, Tsing F. (*J* 1933), Htg. Engr., Andersen, Meyer & Co., Ltd., and (for mail), No. 16, Lane 152, Edinburgh Rd., Shanghai, China.

WEBB, John S. (*M* 1920), 16 Brookline St., Needham, Mass.

WEBB, John W. (*M* 1926), Managing Dir., Webb Dust Removing & Drying Co., Ltd., Princess St. Works, and (for mail), 6 Meadows Rd., Heaton Chapel, Stockport, England.

WEBSTER, E. Kessler (*M* 1915), Warren Webster & Co., 17th and Federal Sts., Camden, N. J.

WEBSTER, Warren (*Life Member*; *M* 1906; *A* 1899), Warren Webster & Co., 17th and Federal Sts., Camden, N. J.

WEBSTER, Warren, Jr. (*M* 1932; *A* 1932; *J* 1927), Vice-Pres-Treas. (for mail), Warren Webster & Co., 17th and Federal Sts., Camden, and Washington Ave. and Colonial Ridge, Haddonfield, N. J.

WECHSBERG, Otto (*M* 1932), Pres. and Gen. Mgr., Coppus Engrg. Corp., 344 Park Ave., and (for mail), 1006 Main St., Worcester, Mass.

WEGMANN, Albert (*M* 1918), 6206 North 17th St., Philadelphia, Pa.

WEIL, Martin (*A* 1925), Vice-Pres. (for mail), Weil-McLain Co., 641 W. Lake St., and 4259 Hazel Ave., Chicago, Ill.

WEIL, Maurice I. (*A* 1928), Pres. (for mail), Chicago Pump Co., 2336 Wolfram St., and 1409 Elmdale Ave., Chicago, Ill.

WEIMER, Fred G. (*A* 1919), Salesman, Kewanee Boiler Corp., 1741 W. St. Paul Ave., and (for mail), 3958 N. Stowell Ave., Milwaukee, Wis.

WEINSHANK, Theodore* (*Life Member*; *M* 1906), (Board of Governors, 1913), 3307 Belden Ave., Chicago, Ill.

WEISS, Arthur P. (*M* 1928), 134 Farrington Ave., North Tarrytown, N. Y.

WEISS, Carl A. (*A* 1924), Supt. (for mail), Kornbrodt Kornice Ko., 1811 Troost Ave., and 29 East 68th St., Kansas City, Mo.

WEITZEL, Paul H. (*S* 1934), Cameron B. Weitzel, and (for mail), 122 E. High St., Manheim, Pa.

WELCH, Louis A., Jr. (*A* 1929), 443 Second St., Schenectady, N. Y.

WELDY, Lloyd O. (*M* 1930), Sales Engr. (for mail), Powers Regulator Co., 2720 Greenview Ave., and 2846 North 77th Ave., Chicago, Ill.

WELSH, Harry S. (*M* 1906), Sales Engr., Weil-McLain Co., and (for mail), 53 Kemphurst Rd., Rochester, N. Y.

WELTER, M. A. (*A* 1925) (for mail), Twin City Furnace Co., 410-12 W. Lake St., and 4306 S. Garfield, Minneapolis, Minn.

WENDT, Edgar F. (*M* 1918), Pres. (for mail), Buffalo Forge Co., 490 Broadway, and 120 Lincoln Pkwy., Buffalo, N. Y.

WEST, Perry* (*M* 1911), (Council, 1920-1925; Treas., 1924-1925), Consulting Engr. (for mail), 13 Central Ave., and 445 Ridge St., Newark, N. J.

WETZELL, Horace E. (*M* 1934), Chief Engr. (for mail), The Smith & Oby Co., 6107 Carnegie Ave., and 8796 Elsmere Dr., Cleveland, Ohio.

WHALLON, Fletcher (*S* 1935), 3852 Lyndale Ave. S., Minneapolis, Minn.

WHEELER, Otto J. (*M* 1923), Pres-Treas. (for mail), The Samuel A. Esswein Htg. & Plbg. Co., 548-558 W. Broad St., and 2044 Collingswood Rd., Columbus, Ohio.

WHELLER, Harry S. (*M* 1916), Vice-Pres., L. J. Wing Mfg. Co., 154 West 14th St., New York, and (for mail), 725 Union Ave., Elizabeth, N. J.

WHITE, Eugene B. (*M* 1934), Architect and Engr. (for mail), Y. M. C. A., 19 S. LaSalle St., Chicago, and 309 N. Taylor Ave., Oak Park, Ill.

WHITE, Everett A. (*M* 1921), Engrg. Dept., Crane Co., 30 South 16th St., and (for mail), 5244 Nottingham St., St. Louis, Mo.

WHITE, Elwood S. (*M* 1921), Pres. (for mail), Taco Heaters, Inc., Room 1224, 342 Madison Ave., New York, N. Y., and Meadowbank Rd., Old Greenwich, Conn.

WHITE, John C. (*M* 1932), State Power Plant Engr. (for mail), 624 E. Main St., and 622 E. Main St. Madison, Wis.

WHITELAW, H. Leigh (*M* 1916), Vice-Pres. (for mail), American Gas Products Corp., 40 West 40th St., New York, N. Y., and Overbrook Lane, Darien, Conn.

WHITELEY, Stockett M. (*M* 1933), Consulting Engr. (for mail), Baltimore Life Bldg., and 3931 Canterbury Rd., Baltimore, Md.

WHITMER, Robert P. (*M* 1935), Secy., American Foundry & Furnace Co., and (for mail), 1402 E. Washington St., Bloomington, Ill.

WHITSON, Lee S. (*S* 1935), 4841 Harriet Ave., Minneapolis, Minn.

WHITTALL, Ernest T. (*A* 1933), Vice-Pres. (for mail), May Oil Burner of Canada, Ltd., 196 Adelaide St. W., and 11 Cottingham Rd., Toronto, Ont., Canada.

WIEGNER, Henry B. (*M* 1919), Mgr., Boston Office, Johnson Service Co., 20 Winchester St., Boston, and (for mail), 143 Standish Rd., Watertown, Mass.

WIERENGA, Peter O. (*A* 1931), Vice-Pres. (for mail), C. C. James Co., 49 Coldbrook St. N.E., and 231 Brown St. S.E., Grand Rapids, Mich.

WIGGINS, Oswald James (*J* 1935; *S* 1933), Walnut Grove, Minn.

WIGGS, G. Lorne (*A* 1932; *J* 1924), Consulting Engr. (for mail), University Tower, and 4797 Grosvenor Ave., Montreal, Que., Canada.

WIGLE, Bruce M. (*A* 1926), Pres. (for mail), Bruce Wigle Plbg. & Htg. Co., 9117 Hamilton Ave., and 18114 Oak Dr., Detroit, Mich.

WILDER, Edward L. (*M* 1915), Mgr., Gas Sales (for mail), Utility Management Corp., 120 Wall St., New York, and 149 Mt. Joy Place, New Rochelle, N. Y.

WILEY, Edgar C. (*M* 1909), Wiley & Wilson, Lynchburg, Va.

WILHELM, Joseph E. (*S* 1934), 1355 West 87th, Cleveland, Ohio.

WILKINSON, Farley J. (*M* 1933), Engr., Montgomery Ward & Co., Chicago, and (for mail), 18257 Martin Ave., Homewood, Ill.

WILLARD, Arthur Cutts* (*M* 1914), (*Presidential Member*), (Pres., 1928; 1st Vice-Pres., 1927; 2nd Vice-Pres., 1926; Council, 1925-1929), Pres. (for mail), University of Illinois, President's Office, and 711 Florida Ave., Urbana, Ill.

WILLEY, Earl C. (*M* 1934), Mech. Engrg. Instructor, Oregon State College, and (for mail), 1652 "A" St., Corvallis, Ore.

WILLIAMS, Allen W. (*A* 1915), Managing Director (for mail), National Warm Air Htg. & Air Conditioning Assn., 50 W. Broad St., Columbus, and 51 Meadow Park Ave., Boxley, Ohio.

WILLIAMS, Frank H. (*J* 1934), Air Cond. Tester, Frigidaire Div., General Motors, Frigidaire Corp., and (for mail), 14 Grand Apts., Dayton, Ohio.

WILLIAMS, J. McFarland, Jr. (*A* 1928; *J* 1927), Sales Engr., 1407-35th St. N.W., Washington, D. C.

WILLIAMS, J. Walter (*M* 1915), Pres. (for mail), Forest City Plbg. Co., 332-36 E. State St., and 923 E. State St., Ithaca, N. Y.

WILLIAMS, Leo E. (*A* 1933; *J* 1930), Viscose Co., and (for mail), 827 Liberty St., Meadville, Pa.

WILLIS, Leonard L. (*S* 1935), 1212 Oliver N., Minneapolis, Minn.

WILMOT, Charles S. (*M* 1919), (for mail), 106 South 16th St., Philadelphia, and 406 Essex Ave., Narberth, Pa.

WILSON, George T. (*M* 1925), Sales Engr., Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto, and (for mail), Tyre Ave., Islington, Ont., Canada.

WILSON, Harold A., Jr. (*J* 1933), General Sales Dept., American Radiator Co., 40 West 40th St., and (for mail), 1133 Park Ave., New York, N. Y.

WILSON, Raymond W. (*M* 1934), Member of Firm (for mail), Wilson-Brinker Co., 412 Pythian Bldg., and 429 Creston Ave., Kalamazoo, Mich.

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WITMER, Charles N. (*J* 1930), Dist. Dealer Supv. (for mail), Carrier Engrg. Corp., 2022 Bryan St., and 4154½ Prescott St., Dallas, Texas.

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WOHL, Maurice W. (*M* 1934), Engr., American Insulating Corp., 377 Atlantic Ave., and (for mail), 32 Lenox Rd., Brooklyn, N. Y.

WOLF, J. C. (*M* 1923), Drafting Room Engr. (for mail), B. F. Sturtevant Co., and 44 Central Ave., Hyde Park, Mass.

WOOD, Frederick C. (*J* 1931), Sales Air Cond. Engr. (for mail), Westerlin & Campbell Co. (Agents York Ice Machinery Corp.), 1113 Cornelia Ave., and 1905 Estes Ave., Chicago, Ill.

WOOD, J. Sydney (*M* 1926), Estimator (for mail), Bennett & Wright, Ltd., 72 Queen St. E., and 163 Briar Hill Ave., Toronto, Ont., Canada.

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WOODS, Edward H. (*M* 1934), Prop. (for mail), F. H. Higgins, 311 E. State St., and Hook Pl., Ithaca, N. Y.

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Y

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Z

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Summary of Membership

(Corrected to January 1, 1935)

UNITED STATES

Alabama	1	Montana	3
Arizona	3	Nebraska	2
Arkansas	3	New Jersey	100
California	34	New York	357
Colorado	5	North Carolina	7
Connecticut	30	North Dakota	1
Delaware	5	Ohio	101
District of Columbia	16	Oklahoma	27
Florida	3	Oregon	1
Georgia	6	Pennsylvania	210
Illinois	195	Rhode Island	5
Indiana	18	South Carolina	2
Iowa	6	Tennessee	6
Kansas	2	Texas	21
Kentucky	6	Utah	1
Louisiana	3	Vermont	3
Maine	4	Virginia	12
Maryland	21	Washington	24
Massachusetts	103	West Virginia	5
Michigan	94	Wisconsin	50
Minnesota	117		
Missouri	81		1694

FOREIGN COUNTRIES

Australia	3	Japan	5
Belgium	1	Mexico	3
Canada	89	New Zealand	3
China	9	Norway	2
Czechoslovakia	1	Phillipine Islands	1
Denmark	1	Scotland	1
England	16	South Africa	1
France	8	Sweden	3
Germany	2	U. S. S. R.	1
Holland	1		
India	1		156
Ireland	1		
Italy	3	Total Membership	1850

SUMMARY OF MEMBERSHIP BY GRADES

Honorary Members	2
Presidential Members	23
Members	1156
Associate Members	339
Junior Members	233
Student Members	97
	1850

LIST OF MEMBERS
Geographically Arranged

UNITED STATES

ALABAMA

Birmingham—
Lichty, C. P.

ARIZONA

Tucson—
Moreau, D.

Phoenix—
Keys, L. F.

Lowell—
Vinson, N. L.

ARKANSAS

Fort Smith—
Herrick, L.

Pine Bluff—
Greer, W. R.

Siloam Springs—
Jones, C. R.

CALIFORNIA

Berkeley—
Duncan, G. W., Jr.

Beverly Hills—
Nelson, H. A.

Huntington Park—
Barnum, W. E., Jr.

Los Angeles—
Anderson, C. S.
Binford, W. M.
Bullock, H. H.
Cranston, W. E., Jr.
Ellingwood, E. L.
Hill, F. M.
Holliday, W. L.
Hungerford, L.
Kendall, E. H.
Kennedy, M.
Kooistra, J. F.
Ness, W. H. C.
Orear, A. G.
Ott, O. W.
Park, J. F.
Pierce, E. D.
Polderman, L. H.
Scofield, P. C.
Simonds, A. H.

Oakland—
Cummings, G. J.

Pasadena—
Gifford, R. L.

San Diego
Sadler, C. B.

San Francisco—
Bouey, A. J.
Cochran, L. H.
Corrao, J.
Haley, H. S.
Hudson, R. A.
Krueger, J. I.
Leland, W. E.
Scott, W. P., Jr.

South Pasadena—
Warren, H. L.

COLORADO

Colorado Springs—
Davis, A. F.
Jardine, D. C.

Denver—
McQuaid, D. J.
O'Rear, L. R.
Ward, O. G.

CONNECTICUT

Bridgeport—
Faile, E. H.
Smak, J. R.

Fairfield—
Osborn, W. J.

Greenwich—
Jones, A. L.
Opperman, E. F.

Manchester—
Buck, L.
Millard, J. W.

New Britain—
Hjerpe, C. A., Jr.

New Haven—
Greenburg, L.
Hughes, C. E.
Seeley, L. E.
Teasdale, L. A.
Winslow, C.-E. A.

New London—
Chapin, C. G.
Forsberg, W.
Hopson, W. T.

Norwalk—
Mead, E. A.

Riverside—
Murphy, J. R.

South Norwalk—
Adams, H. E.
Harvey, A. D.
Jennings, I. C.
Lyons, C. J.
Wylie, H. M.

Stamford—
Hoyt, L. W.

Torrington—
Doster, A.

Waterbury—
Ahlberg, H. B.
Ahlff, A. A.
Hutzel, H. F.
Simpson, W. K.
Stewart, C. W.

DELAWARE

Wilmington—
Gawthrop, F. H.
Hayman, A. E., Jr.
Kershaw, M. G.
Lownsberry, B. F.
Schoenijahn, R. P.

DISTRICT OF COLUMBIA

Washington—
Beitzell, A. E.
Brown, W. A.
Coward, H.
Downes, H. H.
Febrey, E. J.
Feltwell, R. H.
Frankel, G. S.
Gardner, S. F.
Hood, O. P.
Kiczales, M. D.
Mayette, C. E.
Ourusoff, L. S.
Smallman, E. W.
Thompson, N. S.
Urdahl, T. H.
Williams, J. M., Jr.

FLORIDA

Ft. Lauderdale—
Charlton, J. F.

West Palm Beach—
Hodeaux, W. L.
Pizie, S. G.

GEORGIA

Atlanta—
Clare, F. W.
Kent, L. F.
Klein, E. W.
McKinney, W. J.
Templin, C. L.
Thornton, W. B.

ILLINOIS

Bloomington—
MaGirl, W. J.
Soper, H. A.
Whitmer, R. P.

Chicago—
Aeberly, J. J.
Arenberg, M. K.
Baumgardner, C. M.
Black, F. C.
Bolte, E. E.
Borling, J. R.
Bracken, J. H.
Braun, L. T.
Broom, B. A.
Brown, A. P.
Brown, T.
Carman, G. G.
Casey, B. L.
Christman, W. F.
Close, P. D.
Crone, C. E., Jr.
Cunningham, T. M.
Cutler, J. A.
Dean, F. J., Jr.
DeLand, C. W.
Doherty, R.
Dunham, C. A.
Emmert, L. D.
Ericsson, E. B.
Evleth, E. B.
Finan, J. J.
Fitzgerald, M. J.
Fleming, J. P.
Foster, T. R.
Frank, J. M.
Gardner, W., Jr.
Gaylord, F. H.
Getschow, G. M.
Getschow, R. M.
Gibbs, F. C.
Goelz, A. H.
Gossett, E. J.
Graves, W. B.
Haas, S. L.
Haines, J. J.
Hale, J. F.
Hanley, T. F., Jr.
Hart, H. M.
Hattis, R. E.
Hayden, C. F.
Hayes, J. J.
Hayward, R. B.
Heckel, E. P.
Herlihy, J. J.
Hill, E. V.
Hinckley, H. B.
Hornung, J. C.
Horton, H. F.
Howatt, J.
Howell, L.
Hubbard, G. W.
Hustoel, A. M.
Jennings, W. G.
Jenson, J. S.
Johns, H. B.
Johnson, C. W.
Keeney, F. P.

Kehm, H. S.
Kreissl, H. G.
Lagodzinski, H. J.
Lautenschlager, F.
Lawler, M. M.
Lees, H. K.
Lewis, S. R.
Lockhart, H. A.
Machen, J. T.
Malone, D. G.
Malvin, R. C.
Marschall, P. J.
Martin, A. B.
Matchett, J. C.
Mathis, E.
Mathis, H.
Mathis, J. W.
Mathis, V. J.
Mauer, W. J.
May, M. F.
McCauley, J. H.
McClellan, J. E.
McDonnell, E. N.
McFarland, W. P.
McIlvaine, J. H.
Mertz, W. A.
Miller, F. A.
Miller, J. E.
Miller, R. T.
Milliken, J. H.
Mittendorff, E. M.
Mueller, H. C.
Murphy, E. T.
Narowetz, L. L., Jr.
Needler, J. H.
Neiler, S. G.
Nelson, C. L.
Newport, C. F.
O'Brien, J. H.
Offen, B.
Olsen, C. F.
Peller, L.
Pitcher, L. J.
Pope, S. A.
Powers, F. W.
Prentice, O. J.
Price, C. E.
Rasmussen, R. P.
Raymond, F. I.
Reger, H. P.
Reid, H. P.
Ries, L. S.
Rietz, E. W.
Rottmayer, S. I.
Sawhill, R. V.
Scheidecker, D. B.
Schweim, H. J.
Seelig, L.
Shultz, E.
Small, J. D.
Snider, L. A.
Spielman, G. P.
Stannard, J. M.
Sutcliffe, A. G.
Thinn, C. A.
Thomas, L. G. L.
Thomas, R. H.
Thommen, A. A.
Tobin, J. F.
Tornquist, E. L.
Truitt, J. E.
Trumbo, S. M.
Turner, G. G.
Vernon, J. R.
Wallace, K. S.
Walters, W. T.
Walther, V. H.
Warren, F. C.
Washington, G.
Washington, L. W.
Weil, M.
Weil, M. I.
Weinshank, T.
Weldy, L. O.
White, E. B.
Wilson, W. H.
Wood, F. C.
Zack, H. J.
Zimmerman, A. H.

Cicero—
Keppner, H. W.

Decatur—
Shorb, W. A.

Elmhurst—
Potvin, L. J.

Evanston—
Hayes, J. J.
Moore, R. E.

Homewood—
Wilkinson, F. J.

Joliet—
Russell, W. B.

Kewanee—
Bronson, C. E.
Dickson, R. B.
Hartman, J. M.
Pursell, H. E.

LaGrange—
Eaton, B. K.
Linn, H. R.

Moline—
Beling, E. H.
Nelson, H. W.
Nelson, R. H.
Otis, G. E.

Mt. Vernon—
Benoist, L. L.

Norwood Park—
Olson, B.

Oak Park—
Blanding, G. H.
May, E. M.
Nightingale, G. F.
Uhlhorn, W. J.

Peoria—
Farnsworth, J. G.
Meyer, F. L.
Robb, J. M.

Rockford—
Braatz, C. J.
Dewey, R. P.
Merwin, G. E.
Sharp, H. C.
Stewart, D. J.

Urbana—
Broderick, E. L.
Fahnestock, M. K.
Konzo, S.
Kratz, A. P.
Severns, W. H.
Willard, A. C.
Woodruff, W. J.

Villa Park—
Armspach, O. W.

Wilmette—
Norris, W. D.

Winnetka—
Ellis, E. E.

INDIANA

Evansville—
Bulleit, C. R.

Huntington—
Smith, G. W.

Indianapolis—
Ammerman, C. R.
Fenstermaker, S. E.
Hagedon, C. H.
Hayes, J. G.
Kruse, R. W.
Ott, R. C.
Poehner, R. E.
Shivers, P. F.
Supple, G. B.
Warren, C. N.

Lafayette—
Hoffman, J. D.

LaPorte—
Shrock, J. H.

Michigan City—
Stockwell, W. R.

Peru—
Pyle, J. W.
Thrush, H. A.

St. Mary-of-the-Woods
Bisch, B. J.

IOWA

Ackley—
Nelson, G. O.

Cedar Rapids—
Chandler, C. W.

Council Bluffs—
Huffacker, H. B.

LeMars—
Mathey, N. J.

Sioux City—
Hagan, U. V.

Waterloo—
Winterbottom, R. F.

KANSAS

Hutchinson—
Stevens, H. L.

Salina—
Ryan, W. F.

KENTUCKY

Cold Spring—
Ward, F. J.

Ft. Thomas—
Stevens, W. R.

Lexington—
O'Bannon, L. S.

Louisville—
Hellstrom, J.
Murphy, H. C.
Reed, W. M.

LOUISIANA

Choudrant—
Rinehart, W. R.

New Orleans—
Gammill, O. E., Jr.
May, G. E.

MAINE

Auburn—
Fowles, H. H.

Portland—
Fels, A. B.
Merrill, C. J.
Swanson, H.

MARYLAND

Baltimore—
Axeman, J. E.
Collier, W. I.
Hall, M. S.
Lednum, J. M.
Leilich, R. L.
Maccubbin, H. A.
McCormack, D.
Morris, E. J.
Posey, J.
Seiter, J. E.
Shepard, J. deB.
Smoot, T. H.
Vance, L. G.
Vincent, P. J.
Walsh, J. L.
Whiteley, S. M.

Brooklyn Park—
Rodgers, J. S.

Chevy Chase—
Dalla Valle, J. M.

Ellicott City—
Tibbets, J. C.

Rockville—
Brunett, A. L.

Roland Park—
Dorsey, F. C.

MASSACHUSETTS

Arlington—
Shaw, N. J. H.

Arlington Heights—
Tarr, H. M.

Boston—
Archer, D. M.
Bartlett, A. C.
Berchtold, E. W.
Boyden, D. S.
Brinton, J. W.
Brissette, L. A.
Bryant, A. G.
Bullock, T. A.
Cummings, C. H.
Drinker, P.
Dusossoit, E. A.
Edwards, D. J.
Foulds, P. A. L.
Franklin, R. S.
Gleason, G. H.
Hajek, W. J.
Herrick, D. A.
Hilliard, C. E.

Hoyt, C. W.
Keefe, E. T.
Kelley, J. J.
Kellogg, A.
Kimball, C. W.
McCoy, T. F.
Merrill, F. A.
Miller, J. F. G.
Moulton, D.
Osborne, M. M.
Plunkett, J. H.
Rydell, C. A.
Shaw, E.
Stetson, L. R.
Swaney, C. R.
Turner, J.
Tuttle, J. F.
Waterman, J. H.
Yaglou, C. P.

Brookline—
Riley, E. C.

Cambridge—
Baker, R. H.
Flint, C. T.
Haddock, I. T.
Holt, J. W.
Kahan, C.
Lees, J. T.
MacDonald, E. A.

Cochituate—
Ahearn, W. J.

Dalton—
Dakin, H. W.

E. Dedham—
Higgins, T. J.

Dorchester—
Brown, M.
Goodrich, C. F.
Hosterman, C. O.
Shaer, I. E.

Fitchburg—
Karlson, A. F.
McKittrick, P. A.

Harwich Port—
Maxwell, G. W.

Holbrook—
Nason, G. L.

Hyde Park—
Ellis, F. R.
Epple, A. B.
Fritzberg, L. H.
Keyes, R. E.
Wolf, J. C.

Ipswich—
Monroe, R. R.

Lawrence—
Bride, W. T.

Leominster—
Kern, R. T.

Lexington—
Brigham, F. H.

Lynn—
Feehan, J. B.
Oates, W. A.

Medford—
Cushman, L. D.

Melrose—
Cole, E. Q.
Dodge, H. G.
Gerrish, G. B.

Milton—
Mitchell, C. H.

Needham—
Park, C. D.
Webb, J. S.

Newton Center—
Murray, J. J.

Newtonville—
Emerson, R. R.
McMurrer, L. J.

Quincy—
Gesmer, J.
Stone, E. R.

Reading—
Ingalls, F. D. B.

Revere—
Foulds, S. T. N.

South Hadley—
Colby, C. W.

Southwick—
Shaw, B. E.

Springfield—
Brown, W. M.
Cross, R. E.
Holmes, R. E.
Leland, W. B.
Murphy, W. W.
O'Neil, J. M.

Waban—
Jones, W. T.

Watertown—
Wiegner, H. B.

Wellesley Hills—
Barnes, W. E.
Gilling, W. F., Jr.

West Roxbury—
Christie, A. Y.
McPherson, W. A.

Weymouth—
Clough, L.

Winchester—
Jackson, A. B.

Woburn—
Parker, P.

Wollaston—
Hodgdon, H. A.

Worcester—
Robinson, H. C.
Wechsberg, O.

MICHIGAN

Ann Arbor—
Backus, T. H. L.

Battle Creek—
Christenson, H.

Birmingham—
Hadjisky, J. N.

Detroit—
Akers, G. W.
Arnoldy, W. F.
Baldwin, W. H.
Barth, H. E.
Bishop, F. R.
Blackmore, F. H.
Boales, W. G.
Booth, H. N.
Brennan, J. W.
Chappell, H. D.
Chester, T.
Collamore, R.
Connell, R. F.
Coon, T. E.
Cooper, F. D.
Cummins, G. H.
Darlington, A. P.
Dauch, E. O.
Dubry, E. E.
Eggleston, L. W.
Feely, F. J.
Giguere, G. H.
Glanz, E.
Hamlin, H. A.
Hare, W. A.
Harms, W. T.
Harrigan, E. M.
Heydon, C. G.
Hogan, E. L.
Kilner, J. S.
Kirkpatrick, A. H.
Knibb, A. E.
Luty, D. J.
Maier, G. M.
McColl, J. R.
McConachie, L. L.
McIntire, J. F.
McLean, D.
Milward, R. K.
Morse, C. T.
Olson, R. G.
Paetz, H. E.
Parrott, L. G.
Partlan, J. W.
Purcell, F. C.
Purcell, R. E.
Randall, W. C.
Rowe, W. A.
Sanford, S. S.
Sauer, R. L.
Schlichting, W. G.
Shea, M. B.
Snell, E.
Snyder, J. W.
Spitzley, R. L.
Spurgeon, J. H.
Taylor, W. E.
Taze, D. L.
Toonder, C. L.
Turner, J. W.
Tuttle, G. H.
Walker, J. H.
Wallace, J. B.
Wallich, A. C.
Wigle, B. M.
Winans, G. D.

Dowagiac—
Firestone, J. F.
Torr, T. W.

E. Lansing—
Miller, L. G.

Grand Rapids—
Bradfield, W. W.
Leigh, R. L.
Morton, C. H.
Troske, J. J.
Wierenga, P. O.
Ziesse, K. L.

Grosse Point Park—
Vogt, J. B.

Holland—
Cherven, V. W.
Sawyer, J. N.
Van Alsburg, J. H.

Houghton—
Seeber, R. R.

Kalamazoo—
Brinker, H. A.
Downs, S. H.
McConner, C. R.
Temple, W. J.
Wilson, R. W.

Lansing—
Adams, E. I.
Distel, F.
McLouth, B. F.
Parsons, R. A.

Muskegon Heights—
Reid, H. F.

St. Joseph—
Milliken, V. D.

MINNESOTA

Cloquet—
Spafford, A.

Duluth—
Foster, C..

Minneapolis—
Aiken, J. F.
Algren, A. B.
Anderson, D. B.
Anderson, S. H.
Armstrong, R. W.
Bell, E. F.
Betts, H. M.
Bjerken, M. H.
Bredesen, B. P.
Bull, A. S.
Buot, A. V.
Burns, E. J.
Burritt, C. G.
Butts, R. L.
Cash, T. T.
Copperud, E. R.
Dahlstrom, G. A.
Dovolis, N. J.
Forfar, D. M.
Gelb, A.
Gerrish, H. E.
Gordon, E. B., Jr.
Gordon, W. J., Jr.
Gross, L. C.
Haatvedt, S. R.
Hall, J. R.
Hanson, L. P.
Harris, J. B.
Hildebrandt, H. A.
Hitchcock, P. C.
Howard, E.
Huch, A. J.
Johnson, L. H.
Jones, N. W.
Jordan, L. E.
Jordan, R. C.
King, R. L.
Knudtson, C. M.
Kuehn, W. C.
Kuempel, L. L.
Kuns, J. W.
Lange, F. F.
Legler, F. W.
Leslie, D. E.
Lewis, C. E.
Lowe, H. H.
Lund, C. E.
Lupient, G. C.

Magney, G. R.
Martenis, J. V.
Maynard, H. R.
Miller, H. A.
Miller, L. B.
Mjolsnes, L. O.
Morgan, G. C.
Morton, H. S.
Myers, C. R.
Nelson, R. A.
Noble, T. G.
Ostrin, A.
Pappenfus, W. G.
Pfeifer, O. J., Jr.
Porter, H. M.
Powell, K. A.
Priester, G. B.
Pung, D. W.
Roberts, J. R.
Rossiter, P. A.
Rowley, F. B.
Sanford, A. L.
Schernbeck, F. H.
Steinkellner, E. J.
Stiller, F. W.
Sundell, S. S.
Sutherland, D. L.
Swanson, R. G.
Sweiven, C. E.
Swenson, J. E.
Tabor, C. B.
Uhl, E. J.
Uhl, W. F.
Van Horn, H. T.
Welter, M. A.
Whallon, F.
Whitson, L. S.
Willis, L. L.

Owatonna—
Clarkson, W. B.

Robbinsdale—
Sweatt, C. H.

Rochester—
Adams, N. D.

Shevlin—
Nesdahl, E.

St. Paul—
Arnold, E. Y.
Backstrom, R. E.
Barnum, C. R.
Buenger, A.
Fitts, C. D.
Gausman, C. E.
Gill, J. W.
Haslett, H. M.
Hickey, D. W.
Hyde, L. L.
Jones, E. F.
Kolinsky, M. D.
Lindberg, A. F.
McNamara, W.
Oberg, H. C.
Pennel, R.
Poucher, R. C.
Ruff, D. C.
Shaw, H. W.
Swanstrom, A. E.
Winterer, F. C.
Wunderlich, M. S.

Walnut Grove—
Wiggins, O. J.

Wayzata—
Heberling, C. W.

Winona—
Hamerski, F. D

MISSOURI

Ferguson—
Szombathy, L. R.

Jefferson City—
Woodman, L. E.

Kansas City—
Adams, C. W.
Allan, N. J.
Arthur, J. M.
Betz, H. D.
Bliss, G. L.
Buckley, M. B.
Caleb, D.
Campbell, E. K.
Campbell, E. K., Jr.
Chase, L. R.
Clegg, C.
Cook, B. F.
Dawson, T. L.
Disney, M. A.
Dodds, F. F.
Downes, N. W.
Eppright, J. O.
Fehlig, J. B.
Filkins, H. L.
Flarsheim, C. A.
Gillham, W. E.
Haas, E., Jr.
Kell, W. R.
Kitchen, J. H.
Lewis, J. G.
Maillard, A. L.
Matthews, J. E.
Millis, L. W.
Natkin, B.
Nottberg, G.
Nottberg, H.
Olchoff, M.
Olson, G. E.
Pines, S.
Rudio, H. M.
Russell, W. A.
Sheppard, F. A.
Stephenson, L. A.
Weiss, C. A.
Wymore, F. C.
Zink, D. D.

Maplewood—
Walters, A. L.

St. Louis—
Barry, J. G., Jr.
Bayse, H. V.
Bradley, E. P.
Carlson, E. E.
Clegg, R. R.
Cooper, J. W.
Corrigan, J. A.
Davis, C. R.
DuBois, L. J.
Edwards, D. F.
Fagin, D. J.
Falvey, J. D.
Fillo, F. B.
Foster, J. M.
Gilmore, L. A.
Griffin, J. J.
Grossmann, H. A.
Hamilton, J. E.
Hartwein, C. E.
Hester, T. J.
Kent, J. K.
Langenberg, E. B.
Manahan, J. E.
McLeary, H. W.
McMahon, T. W.
Moon, L. W.
Reed, P. L.
Rodenheiser, G. B.
Rosebrough, R. M.
Sodemann, P. W.
Sodemann, W. C. B.
Stammer, E. L.
Tenkonohy, R. J.
White, E. A.

Webster Groves—
Myers, G. W. F.
Ranck, G. L.

Wellston—
Seepe, P. E.

MONTANA

Big Timber—
Strickland, A. W.

Billings—
Cohagen, C. C.

Bozeman—
Powers, F. I.

NEBRASKA

Clarks—
Manning, W. M.

Scotts Bluff—
Davis, O. E.

NEW JERSEY

Arlington—
Adler, A. A.
Bock, B. A.
Emery, H.

Asbury Park—
Strevell, R. P.

Atlantic City—
Strouse, S. B.

Audubon—
Sanbern, E. N.

Belmar—
Merkel, F. P.

Bloomfield—
Hochuli, H. W.
Tenney, D.

Camden—
Brown, W. M.
Kappel, G. W. A.
Lanning, E. K.
Webster, E. K.
Webster, W.
Webster, W., Jr.

Collingswood—
Plum, L. H.

East Orange—
Atkinson, K. B.
Ferguson, R. R.
Gombers, H. B.
Grahn, V. F.
Rack, E. C.
Reilly, J. H.
Tallmadge, W.
Turno, W. G. W.
Worsham, H.

Elizabeth—
Burke, J. J.
Cherne, R. E.
Cornwall, G. I.
Grant, W. A.
Lyman, S. E.
Merle, A.
Wheller, H. S.

Essex Fells—
Stacey, A. E., Jr.

Grantwood—
Butler, P. D.

Hackensack—
Turnau, E. H.

Haddonfield—
Dobbs, C. E.
Jones, R. E.
Moody, L. E.

Hasbrouck Heights—
Goodwin, S. L.

Hawthorne—
Lawton, F. C.

Irvington—
Freas, R. B.
Reinke, A. G.

Jersey City—
Hasagen, J. B.
Jones, H. L.
Kelly, C. J.
Ritchie, W.
Schwartz, J.
Walterthum, J. J.

Kearny—
Holbrook, F. M.

Lyndhurst—
Ehrlich, M. W.

Maplewood—
Evans, W. A.
Kepler, D. A.
Kylberg, V. C.
Smith, M. S.

Merchantville—
Binder, C. G.

Montclair—
Bentz, H

Newark—
Alt, H. L.
Ashley, C. M.
Bryant, P. J.
Carey, P. C.
Carrier, W. H.
Day, V. S.
French, D.
Holton, J. H.
Ingels, M.
Leinroth, J. P.
Lewis, L. L.
Lewis, T.
Lyle, J. I.
Matullo, J. R.
Morehouse, H. P.
Rachal, J. M.
Ray, L. B.
Raymer, W. F., Jr.
Rice, R. B.
Soule, L. C.
Steinmetz, C. W. A.
Tavanlar, E. V.
West, P.

North Arlington—
Bermel, A. H.
Vogelbach, O.

Orange—
Crawford, J. H., Jr.

Paterson—
Cox. H. F.
Pryor, F. L.

Perth Amboy—
Simkin, M.

Plainfield—
Hedges, H. B.
Tobin, G. J.

Ridgefield Park—
Davis, A. C.

Ridgewood—
Fitts, J. C.

Roselle Park—
Kampish, N. S.

South Orange—
Hansen, C. C.

Summit—
Oaks, O. O.

Teaneck—
Heebner, W. M.

Tenafly—
Redfield, C.

Union City—
Taverna, F. F.

Upper Montclair—
Fernald, H. B., Jr.

Verona—
Stone, G. F.

West Orange—
Adlam, T. N.

West New York—
Stinard, R. L.

NEW YORK

Albany—
Anker, G. W.
Bond, H. A.
Johnson, H. S.
Ryan, H. J.
Taggart, R. C.
Teeling, G. A.

Binghamton—
Brown, R. F.
Marum, O.

Bronxville—
Dornheim, G. A.

Buffalo—
Beman, M. C.
Booth, C. A.
Cherry, L. A.
Cheyney, C. C.
Cressy, R. E.
Criqui, A. A.
Currier, C. H.
Danforth, N. L.
Davis, J.
Day, H. C.
Dyer, O. K.
Erdle, G. F.
Evans, C. A.
Farnham, R.
Farrar, C. W.
Gifford, C. A.
Hamlin, C. J., Jr.
Harding, L. A.
Heath, W. R.
Hedley, P. S.
Hexamer, H. D.
Jackson, M. S.
Johnson, E. E.
Kamman, A. R.
Landers, J. J.
Love, C. H.
Madison, R. D.
Mahoney, D. J.
McTernan, F. J.
Mosher, C. H.
Murray, T. F.
Rente, H. W.
Roebuck, W., Jr.
Shelney, T.
Snyder, J. S.
Strouse, S. W.
Thornton, R. T.
Voisinet, W. E.
Walker, E. R.
Wendt, E. F.
Yager, J. J.

Eggertsville—
Eiss, R. M.
Hirschman, W. F.

Elmira—
Davis, B. C.
McGlenn, G. R.

Geneva—
Herendeen, F. W.

Hamburg—
Graham, C. H.

Hastings-on-Hudson—
Reynolds, T. W.

Hudson Falls—
Hollister, E. W.

Irvington-on-Hudson—
Bastedo, A. E.

Ithaca—
Barns, A. A.
Sawdon, W. M.
Williams, J. W.
Woods, E. H.

Jamestown—
Sharp, F. H.

Kendall—
Stangland, B. F.

Kenmore—
Candee, B. C.
Green, J. J.
Roseberry, J. H.

Larchmont—
Gaylor, W. S.

Lockport—
Bishop, C. R.
Saunders, L. P.

Mt. Vernon—
Freitag, F. G.
Northon, L.
Obert, C. W.

New Rochelle—
Abrams, A.
Farley, W. F.
Rose, H. J.

New York City—
Addams, H.
Ames, C. F.
Ashley, E. E.
Atherton, G. R.
Bachler, L. J.
Balsam, C. P. (Brooklyn)
Barbera, H. A.
Barbieri, P. J.
Barnum, M. C.
Baum, A. L.
Beebe, F. E. W.
Bennett, E. A.
Bennitt, G. E.
Berman, L. K.
Bernhard, G (Lawrence, L. I.)
Bernstrom, B.
Bilyeu, W. F.
Blackburn, E. C., Jr. (Hempstead, L. I.)
Blackman, A. O.
Blackmore, J. J.
Blackshaw, J. L. (Brooklyn)
Bodinger, J. H.
Bolton, R. P.
Bowles, P.
Browne, A. L.
Bruckmann, J. C.
Buensod, A. C.
Bulkeley, C. A.
Burbaum, W. A.
Callaghan, P. F., Jr. (Brooklyn)
Callahan, P. J. (Great Kills, S. I.)
Campbell, F. B.
Campbell, R. E.
Carpenter, R. H.
Charles, T. J.
Charlet, L. W.
Chase, C. L. (Brooklyn)
Cohen, N.
Crone, T. E.
Cucci, V. J.
Dailey, J. A. (Astoria, L. I.)
Daly, R. F.
Darts, J. A.
Davison, R. L.
DeBlois, L. A.
Deely, J. J. (Brooklyn)
Denny, H. R.
Donnelly, R.
Downe, E. R.
Driscoll, W. H.
Duff, K.
Durkee, M. E.
Duryea, A. A. (Belden Point, City Island)
Dwyer, T. F. (Brooklyn)
Eadie, J. G.
Eells, H. B. (Brooklyn)
Elliott, L.
Engle, A.
Everetts, J., Jr.
Fansler, P. E.
Fay, F. C.
Fegley, D. R.
Feldman, A. M.
Fenner, N. P.
Ferrero, H. J.
Fiedler, H. W.
Fife, G. D.
Finch, S. B.
Fleisher, W. L.
Flink, C. H.
Frank, O. E.
Frey, G. O.
Friedman, M.
Galloway, J. F. (Brooklyn)
Gautesen, A. (Brooklyn)
Genchi, B. (Brooklyn)
Giannini, A. A.
Gilmour, A. B.
Glore, E. F.
Goldberg, M. (Brooklyn)
Goldschmidt, O. E.
Gordon, P. B.
Gornston, M. H.
Goulding, W. (Brooklyn)
Haigney, J. E. (Brooklyn)
Hamburger, F. G.
Hament, L.
Hartman, F. S.
Hateau, W. M.
Heibel, W. E.
Henry, A. S., Jr.
Herkimer, H.
Herty, F. B. (Brooklyn)
Hertzler, J. R. (Brooklyn)
Hiers, C. R. (Great Neck, L. I.)
Hinkle, E. C. (Hempstead, L. I.)
Hinrichsen, A. F.
Hoffman, C. S.
Hollister, N. A. (Brooklyn)
Hosking, H. L.
Hotchkiss, C. H. B.
Howell, F. B.
Hyman, W. M.
Issertell, H. G.
Jacobus, D. S.
Janet, H. L.
Johnson, E. B. (W. New Brighton, S. I.)
Johnston, W. H.
Kagey, I. B., Jr.
Kastner, G. C.
Kaufman, W. M.
Kenward, S. B. (Bay Shore, L. I.)
Keplinger, W. L.
Kimball, D. D.
Knopf, C.
Kuhlmann, R.
Lane, D. D. (Jackson Heights, L. I.)
Leffingwell, R. R.
Lennon, J. O.
Leupold, H. W.
Lowy, M. R.
Lucke, C. E.
Lyle, E. T.
Lyon, P. S.
Maiman, H. (Glendale, L. I.)
Mandeville, E. W. (Brooklyn)
Marino, D. A.
Markush, E. U.
Marshall, H. H.
Martin, G. W.
McCloughan, C. (Brooklyn)
McKiever, W. H.
McLeish, W. S.

Mehne, C. A.
Meinke, H. G.
Meisel, C. L.
Meyer, C. L.
Meyer, H. C., Jr.
Miller, C. A.
Monroe, M.
Montgomery, O. C.
Moore, R. E.
(Brooklyn)
Morse, F. W.
Moss, E.
(Brooklyn)
Munder, J. F., Jr.
Munier, L. L.
Munro, E. A.
(Lynbrook, L. I.)
Murphy, C. G.
Neale, L. I.
Neary, D. A.
Offner, A. J.
O'Hare, G. W., Jr.
Olsen, G. E.
(Arverne, L. I.)
Olvany, W. J.
Osburn, R. M.
Pabst, C. S.
Patorno, S. A. S.
Pfuhler, J. L.
(W. New Brighton, S. I.)
Phillips, F. W., Jr.
(Brooklyn)
Pihlman, A. A.
Pinder, P. H.
Place, C. R.
Pohle, K. F.
Presdee, C. W.
Price, E. H.
(Riverhead, L. I.)
Purdy, R. B.
Purinton, D. J.
Quigley, W. J.
Quirk, C. H.
Raffes, A.
Raisler, R. K.
Raynis, T.
(Woodhaven L. I.)
Reed, J. F.
Reynolds, W. V.
Richardson, H. T.
Riley, C. L.
Riley, R. C.
(Jamaica, L. I.)
Ritchie, E. J.
Ritter, A.
Rodman, R. W.
Rosenberg, P.
Ross, J. O.
Roth, C. F.
Rozett, W., Jr.
Ruppert, E. H.
Schneider, W. G.
Schoepflin, P. H.
Schulze, B. H.
Scott, C. E.
Scott, G. M.
Scribner, E. D.
Seelbach, H.
Seelig, A. E.
Sellman, N. T.
Senior, R. L.
Seward, P. H.
(Brooklyn)
Shepard, E. C.
Siebs, C. T.
Siegel, L.
(Brooklyn)
Skidmore, J. G.
(Long Island City)
Sklenarik, L.
Stack, F. C.
(Flushing, L. I.)
Staples, W. H.
Sternberg, E.
Sterne, C. M.
(Long Island City)
Stewart, C. W.
Still, F. R.
Strock, C.
Strunin, J.
Sutton, F.
(Babylon, L. I.)
Syska, A. G.
Thomson, T. N.
(Huntington, L. I.)
Tiltz, B. E.
Timmis, W. W.
Tisnower, W.
(Brooklyn)
Torrance, H.
Tucker, F. N.
(Freeport, L. I.)
Tusch, W.
(Brooklyn)
Tyler, R. D.
Vetlesen, G. U.
Vivarttas, E. A.
(Brooklyn)
Vogt, J. H.
Waechter, H. P.
(Thompkinsville, S. I.)
Wachs, L. J.
(Brooklyn)
Wagner, F. H., Jr.
Walker, W. K.
Wallace, G. J.
(Elmhurst, L. I.)
Wallace, W. M., II
(Hollis, L. I.)
Walton, C. W., Jr.
Waring, J. M. S.
White, E. S.
Whitelaw, H. L.
Wilder, E. L.
Wilson, H. A., Jr.
Winquist, W. J.
(Brooklyn)
Wohl, M. W.

Oriskany—
Oakey, W. E.

Ossining—
Hooper, V. F.

Patchoque—
Jalonack, I. G.

Pelham Manor—
Peacock, J. K.

Port Jervis—
Trimmer, C. M.

Rochester—
Betlam, H. T.
Coe, R. T.
Cook, R. P.
Hakes, L. M.
Hutchins, W. H.
Sheldon, N. E.
Stacy, S. C.
Welsh, H. S.

Rome—
Lynch, W. L.
Steele, M. G.

Scarsdale—
Cumming, R. W.
Ullman, H. G.

Schenectady—
Cannon, C. N.
Faust, F. H.
Harrington, E. D.
Hunziker, C. E.
James, J. W.
McLenegan, D. W.
Vogel, A.
Welch, L. A., Jr.

Snyder—
John, V. P.

Syracuse—
Acheson, A. R.
Woese, C. F.

Tarrytown—
Abraham, L.
Weiss, A. P.

Utica—
Steinhorst, T. F.

White Plains—
Johnson, L. O.
Zibold, C. E.

Williamsville—
Rente, S. R.

Yonkers—
Archdeacon, H. K.
Bense, W. M.
Brabbeé, C. W.
Deutchman, J.
Goerg, B.
Hayter, B.
Kelly, J. G.
Rainger, W. F.
Stitt, A. B.
Zuhlke, W. R.

NORTH CAROLINA

Charlotte—
Brandt, E. H.
Christian, C. W.
Hodge, W. B.
Small, B. R.

High Point—
Gray, W. E.

Winston-Salem—
Bahnson, F. F.
Turner, M. E.

NORTH DAKOTA

Grand Forks—
Pesterfield, C. H.

OHIO

Bellefontaine—
Quay, D. M.

Berea—
Curtis, H. F.

Cincinnati—
Bird, C.
Breneman, R. B.
Buford, J. W.
Coombe, J.
Doyle, W. J.
Floyd, M.
Green, W. C.
Helburn, I. B.
Houliston, G. B.
Hust, C. E.
Kiefer, C. J.
Kitchell, H. N.
Kramig, R. E., Jr.
Pillen, H. A.
Pistler, W. C.
Powers, L. G.
Richard, E. J.
Royer, E. B.
Sigmund, R. W.
Smith, J. A.
Sproull, H. E.
Washburn, M. J.
Winther, A.
Wright, K. A.

Cleveland—
Allman, N. S.
Andes, W.
Avery, L. T.
Bailey, E. P., Jr.
Brooks, F. W.
Brueggeman, A. R.
Cohen, P.
Conner, R. M.
Davis, J. R.
Dickenson, F. R.
Eveleth, C. F.
Fonda, B. P.
Fox, O.
Geissbuhler, J. O.
Gottwald, C.
Graham, W. D.
Harvey, L. C.
Hendrickson, J. J.
Kalinsky, A. G.
Kartorie, V. T.
Kitchen, F. A.
Klie, W.
Levy, M. I.
Martinka, P. D.
Matzen, H. B.
O'Gorman, J. S., Jr.
Miles, J. C.
Prendergast, J. J.
Quinlivan, L. P.
Rather, M. F.
Repko, J. J.
Schick, K. W.
Schmidt, R. H.
Schurman, J. A.
Sennet, L. E.
Spencer, R. M.
Steffner, E. F.
Tuve, G. L.
Vanderhoof, A. L.
Wetzell, H. E.
Wilhelm, J. E.
Wise, D. E.

Cleveland Heights—
Davis, R. G.
Rodgers, F. A.

Columbus—
Brown, A. I.
Sherman, R. A.
Slayter, G.
Wheeler, O. J.
Williams, A. W.

Cuyahoga Falls—
Humphrey, D. E.

Dayton—
Hull, H. B.
LaSalvia, J. J.
Moore, H. W.
Williams, F. H.

East Cleveland—
Morris, F. H.
Nobis, H. M.
Read, R. R.
Stark, W. E.

Elyria—
Maynard, J. E.

Lakewood—
Longcoy, G. B.
Ramsey, R. F.
Vermere, E. J.

Mansfield—
Yardley, R. W.

Newark—
Simpson, D. C.

Norwood—
Braun, J. J.
Motz, O. W.

Painesville—
Hobbs, J. C.

Shaker Heights—
Bolz, H. A.
Cary, E. B.

Steubenville—
Smith, R. H.

Toledo—
Baker, H. C.
Myers, F. L.
Treadway, Q.

Youngstown—
Boucherle, H. N.
Choffin, C. C.

OKLAHOMA

Bartlesville—
Yates, G. L.

Norman—
Bowman, J. W.
Chestnutt, N. P.
Cook, A. B.
Dawson, E. F.
Frampton, A. C.
Husky, S. T.
McKinley, C. B.
Rauh, E. M.
Sneed, R. B.
Sonney, K. J.

Oklahoma City—
Beard, E. L.
Dolan, R. G.
Dugger, E. R.
Emmons, N. L.
Gray, E. W.
Howlett, I. G.
Loeffler, F. X.
Loeffler, L., Jr.
Miller, B. R.
Miner, M. H.
Rathbun, P. W.
Rolland, S. L.
Tauson, P. O.
Tiller, L.

Tulsa—
Burke, W. J.
Jones, E.

OREGON

Corvallis—
Willey, E. C.

PENNSYLVANIA

Allentown—
Korn, C. B.

Ardmore—
Haynes, C. V.

Beaver Falls—
Van Alen, W. T.

Beechwood, Del. Co.-
Kipe, J. M.

Bradford—
Black, W. B.

Butler—
Karges, L.

Chambersburg—
Kottcamp, H. A.

Cheltenham—
McElgin, J. W.

Cynwyd—
Smith, W. F.

E. Pittsburgh—
Goodwin, W. C.

Elizabethtown—
Dibble, S. E.

Erie—
Joyce, H. B.

Glenolden—
Grossman, H. E.

Harrisburg—
Eicher, H. C.
Geiger, I. H.
Lutz, J. H., Jr.

Haverford—
Black, E. N., 3rd

Jenkintown—
Slight, I.

Johnstown—
Novotney, T. A.
Stitt, E. W.

Kingston—
MacDonald, D. B.

Lancaster—
Jones, A.
Lloyd, E. C.
Long, D. R.

Lansdowne—
Hansen, C. J.
James, H. R.

Manheim—
Weitzel, P. H.

Marysville—
Gault, G. W.

McKeesport—
Dugan, T. M.

Meadville—
Williams, L. E.

Merion—
Atkins, T. J.

New Castle—
Andrews, G. H.
Sonneborn, C.

Norristown—
Bolsinger, R. C.
Hucker, J. H.

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Tuckerman, G. E.
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Michie, D. F.
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Yates, W.

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Haden, W. N.

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Faber, Dr. O.

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Tyson, W. H.

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Overton, S. H.

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Wm. Kent
J. H. Kinealy
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John Gormly
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Treasurer......Ulysses G. Scollay
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Secretary......Wm. M. Mackay

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R. C. Carpenter
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1911

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1912

President......John R. Allen
1st Vice-President......John F. Hale
2nd Vice-President......Edmund F. Capron
Treasurer......James A. Donnelly
Secretary......Wm. W. Macon

Board of Governors

Chairman, John R. Allen

John F. Hale, *Vice-Chm.*
Edmund F. Capron
R. P. Bolton
Jas. D. Hoffman
Dwight D. Kimball
Samuel R. Lewis
Wm. M. Mackay
Wm. W. Macon, *Secy.*

1913

President......John F. Hale
1st Vice-President......A. B. Franklin
2nd Vice-President......Edmund F. Capron
Treasurer......James A. Donnelly
Secretary......Edwin A. Scott

Board of Governors

Chairman, John F. Hale

A. B. Franklin, *Vice-Chm.*
John R. Allen
Edmund F. Capron
R. P. Bolton
Frank T. Chapman
Ralph Collamore
James A. Donnelly
Dwight D. Kimball
Wm. W. Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President......Samuel R. Lewis
1st Vice-President......Edmund F. Capron
2nd Vice-President......Dwight D. Kimball
Treasurer......James A. Donnelly
Secretary......J. J. Blackmore

Council

Chairman, Samuel R. Lewis

E. F. Capron, *Vice-Chm.*
Dwight D. Kimball
John R. Allen
Frank T. Chapman
Frank I. Cooper
James A. Donnelly
John F. Hale
Harry M. Hart
Frank G. McCann
Wm. W. Macon
James M. Stannard
J. J. Blackmore, *Secy.*

1915

President......Dwight D. Kimball
1st Vice-President......Harry M. Hart
2nd Vice-President......Frank T. Chapman
Treasurer......Homer Addams
Secretary......J. J. Blackmore

Council

Chairman, Dwight D. Kimball

Harry M. Hart, *Vice-Chm.*
Homer Addams
Frank T. Chapman
Frank I. Cooper
E. Vernon Hill
Wm. M. Kingsbury
Samuel R. Lewis
Frank G. McCann
J. T. J. Mellon
Henry C. Meyer, Jr.
Arthur K. Ohmes
J. J. Blackmore, *Secy.*

1916

President......Harry M. Hart
1st Vice-President......Frank T. Chapman
2nd Vice-President......Arthur K. Ohmes
Treasurer......Homer Addams
Secretary......Casin W. Obert

Council

Chairman, Harry M. Hart

F. T. Chapman, *Vice-Chm.*
Homer Addams
Charles R. Bishop
Frank I. Cooper
Milton W. Franklin
E. Vernon Hill
Dwight D. Kimball
Henry C. Meyer, Jr.
Arthur K. Ohmes
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1917

President......J. Irvine Lyle
1st Vice-President......Arthur K. Ohmes
2nd Vice-President......Fred R. Still
Treasurer......Homer Addams
Secretary......Casin W. Obert

Council

Chairman, J. Irvine Lyle

A. K. Ohmes, *Vice-Chm.*
Homer Addams
Davis S. Boyden
Bert C. Davis
Milton W. Franklin
Charles A. Fuller
Harry M. Hart
E. Vernon Hill
James M. Stannard
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1918

President......Fred R. Still
1st Vice-President......Walter S. Timmis
2nd Vice-President......E. Vernon Hill
Treasurer......Homer Addams
Secretary......Casin W. Obert

Council

Chairman, Fred R. Still

W. S. Timmis, *Vice-Chm.*
Homer Addams
William H. Driscoll
Howard H. Fielding
H. P. Gant
C. W. Kimball
J. Irvine Lyle
E. Vernon Hill
Frank G. Phegley
Fred. W. Powers
Champlain L. Riley
Casin W. Obert, *Secy.*

1919

President............Walter S. Timmis
1st Vice-President............E. Vernon Hill
2nd Vice-President............Milton W. Franklin
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Walter S. Timmis

E. Vernon Hill, *Vice-Chm.*
Homer Addams
Howard H. Fielding
Milton W. Franklin
Harry E. Gerrish
George B. Nichols
Frank G. Phegley
Fred. W. Powers
Robt. W. Pryor, Jr.
Champlain L. Riley
Fred R. Still
Casin W. Obert, *Secy.*

1920

President............E. Vernon Hill
1st Vice-President............Champlain L. Riley
2nd Vice-President............Jay R. McColl
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, E. Vernon Hill

C. L. Riley, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Wm. H. Driscoll
A. C. Edgar
Alfred Kellogg
Jay R. McColl
George B. Nichols
Robt. W. Pryor, Jr.
W. S. Timmis
Perry West
Casin W. Obert, *Secy.*

1921

President............Champlain L. Riley
1st Vice-President............Jay R. McColl
2nd Vice-President............H. P. Gant
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Champlain L. Riley

Jay R. McColl, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
H. P. Gant
E. S. Hallett
E. Vernon Hill
Alfred Kellogg
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1922

President............Jay R. McColl
1st Vice-President............H. P. Gant
2nd Vice-President............Samuel E. Dibble
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Jay R. McColl

H. P. Gant, *Vice-Chm.*
Homer Addams
Jos. A. Cutler
Samuel E. Dibble
Wm. H. Driscoll
E. S. Hallett
L. A. Harding
E. E. McNair
H. J. Meyer
C. L. Riley
Perry West
Casin W. Obert, *Secy.*

1923

President............H. P. Gant
1st Vice-President............Homer Addams
2nd Vice-President............E. E. McNair
Treasurer............Wm. H. Driscoll
Secretary............C. W. Obert

Council

Chairman, H. P. Gant

Homer Addams, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
Wm. H. Driscoll
E. S. Hallett
Alfred Kellogg
Thornton Lewis
E. E. McNair
Perry West
Casin W. Obert, *Secy.*

1924

President............Homer Addams
1st Vice-President............S. E. Dibble
2nd Vice-President............William H. Driscoll
Treasurer............Perry West
Secretary............F. C. Houghten

Council

Chairman, Homer Addams

S. E. Dibble, *Vice-Chm.*
F. Paul Anderson
W. H. Carrier
J. A. Cutler
William H. Driscoll
H. P. Gant
W. E. Gillham
L. A. Harding
Alfred Kellogg
Thornton Lewis
Perry West
F. C. Houghten, *Secy.*

1925

President............S. E. Dibble
1st Vice-President............Wm. H. Driscoll
2nd Vice-President............F. Paul Anderson
Treasurer............Perry West
Secretary............F. C. Houghten

Council

Chairman, S. E. Dibble

Wm. H. Driscoll, *Vice-Chm.*
Homer Addams
F. Paul Anderson
W. H. Carrier
J. A. Cutler
W. E. Gillham
W. T. Jones
Thornton Lewis
J. H. Walker
Perry West
A. C. Willard
F. C. Houghten, *Secy.*

1926

President............W. H. Driscoll
1st Vice-President............F. Paul Anderson
2nd Vice-President............A. C. Willard
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson

Council

Chairman, W. H. Driscoll

F. Paul Anderson, *Vice-Chm.*
W. H. Carrier
J. A. Cutler
S. E. Dibble
W. E. Gillham
C. V. Haynes
W. T. Jones
E. B. Langenberg
Thornton Lewis
J. F. McIntire
A. C. Willard

1927

President............F. Paul Anderson
1st Vice-President............A. C. Willard
2nd Vice-President............Thornton Lewis
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson

Council

Chairman, F. Paul Anderson

A. C. Willard, *Vice-Chm.*
H. H. Angus
W. H. Carrier
W. H. Driscoll
Roswell Farnham
H. H. Fielding
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
Thornton Lewis
J. F. McIntire
H. Lee Moore
F. B. Rowley

1928

President............A. C. Willard
1st Vice-President............Thornton Lewis
2nd Vice-President............L. A. Harding
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson

Council

Chairman, A. C. Willard

Thornton Lewis, *Vice-Chm.*
F. Paul Anderson
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
J. F. McIntire
H. Lee Moore
F. B. Rowley

1929

President............Thornton Lewis
1st Vice-President............L. A. Harding
2nd Vice-President............W. H. Carrier
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, Thornton Lewis

L. A. Harding, *Vice-Chm.*
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
W. A. Rowe
F. B. Rowley
A. C. Willard

1930

President............L. A. Harding
1st Vice-President............W. H. Carrier
2nd Vice-President............F. B. Rowley
Treasurer............C. W. Farrar
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, L. A. Harding

W. H. Carrier, *Vice-Chm.*
H. H. Angus
D. S. Boyden
R. H. Carpenter
J. D. Cassell
N. W. Downes
Roswell Farnham
C. W. Farrar
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
Thornton Lewis
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1931

President............W. H. Carrier
1st Vice-President............F. B. Rowley
2nd Vice-President............W. T. Jones
Treasurer............F. D. Mensing
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, W. H. Carrier

F. B. Rowley, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
Roswell Farnham
E. H. Gurney
L. A. Harding
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
F. D. Mensing
W. A. Rowe

1932

President............F. B. Rowley
1st Vice-President............W. T. Jones
2nd Vice-President............C. V. Haynes
Treasurer............F. D. Mensing
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, F. B. Rowley

W. T. Jones, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
W. H. Carrier
John D. Cassell
E. O. Eastwood
Roswell Farnham
F. E. Giesecke
E. Holt Gurney
C. V. Haynes
John Howatt
G. L. Larson
J. F. McIntire
F. D. Mensing
W. E. Stark

1933

President............W. T. Jones
1st Vice-President............C. V. Haynes
2nd Vice-President............John Howatt
Treasurer............D. S. Boyden
Secretary............A. V. Hutchinson

Council

Chairman, W. T. Jones

C. V. Haynes, *Vice-Chm.*
R. H. Carpenter
F. D. Cassell
D. S. Boyden
John Howatt
F. C. McIntosh
L. W. Moon
F. E. Giesecke
G. L. Larson
J. F. McIntire
W. E. Stark
E. K. Campbell
E. O. Eastwood
R. Farnham
E. H. Gurney
F. B. Rowley

1934

President............C. V. Haynes
1st Vice-President............John Howatt
2nd Vice-President............G. L. Larson
Treasurer............D. S. Boyden
Secretary............A. V. Hutchinson

Council

Chairman, C. V. Haynes

John Howatt, *Vice-Chm.*
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
W. T. Jones
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. Walter Moon
O. W. Ott
W. A. Russell
W. E. Stark